DESIGN OF AN AIR-CONDITIONING SYSTEM FOR SECOND AND THIRD FLOORS OF MECHANICAL ENGINEERING DEPARTMENT BUILDING A.B.U.

BY

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DEDICATED

TO MY

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AND

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ABSTRACT

This study has been undertaken to look at some of the air conditioning systems and select one for designing the air conditioning for the second and the third floors of the Mechanical Engineering Department building, Ahmadu Bello University, Zaria. Many types of air conditioning systems are available; an all-air single zone system is selected and designed for the building.

The solar heat gains to the building are calculated using Total Effective Temperature Differential method and the cooling load estimatedrassuming a 12-hour operation of the air conditioning system.

The supply air ducts are sized by the static region whereas the return ducts are sized by velocity reduction method.

A roof mounted air cooled condensing unit (Westings house ST 020) together with a matching air handling unit (Westinghouse AB 060) are proposed to air condition the building.

CHAPTER 1

INTRODUCTION

Air conditioning, of which refrigeration is an inseperable part, has its origins in the fundamental work on thermodynamics which was done by Boyle, Carnot and others, in the seventeenth and eighteenth centuries; but air conditioning as a science applied to practical engineering owes much to the ideas of Carrier at the beginning of this century in the United States.

The design of comfort air conditioning system involves several integrated procedures. For a given problem, the designer must:

a) establish the design conditions
b) estimate the thermal loads
c) select the type of system to be employed
d) calculate the capacity requirement of each component of the system
e) select equipment if commercially available and/or design special equipment
f) design the air distribution system and miscellaneous piping system, and
g) prepare drawings and specifications for the system.

Air conditioning may be definied as

" The process of treating air so as to control simultaneously its temperature, humidity, cleanliness and distribution to meet the requirements of the conditioned space".

Air conditioning systems that are able to perform all air conditioning functions are not in common use. Most do only part of the job. The two most difficult jobs to be performed are satisfactory air cleaning and humidity control.

Most air conditioning systems are only partial air conditioning systems; that is, the system is designed for heating, humidifying, cleaning and distribution, or it is designed for cooling, dehumidifying, cleaning and distribution, as for tropical climates.

Air conditioning of buildings mainly concerns the comfort of people and not maintenance of exact conditions as required for industrial products. The conditions are selected to provide comfort to maximum number of occupants at the same time.

In chapter 2, some air conditioning systems are classified and in later chapters one is selected and designed for the Mechanical Engineering building.

CHAPTER 2

COMFORT AIR CONDITIONING SYSTEMS

The objective of an air conditioning system is to provide a comfortable environment for the occupants of the conditioned space. The comfort environment is the result of simultaneous control of temperature, humidity, cleanliness and air distribution within the occupants vicinity that includes the proper acoustic level.

The final media used to achieve a comfortable environment are air and surfaces surrounding the occupant. Both the air and surfaces of the enclosure are sinks for the metabolic heat evolved by the occupant.

Air conditioning sytems can be broadly divided into:

1) All-air systems
2) Air-water systems
3) All-water systems
4) Unitary and other systems

2.1 All-air systems

An all-air system provides complete sensible heating and cooling, and latent cooling, supplying only air to the conditioned space. The piping between the refrigerating and heat producing devices and the air handling device is generally ignored in this context.

In all-air systems, the air quantity supplied to each terminal must have sufficient capacity to offset both the sensible and latent loads.

All-air systems can be divided into the following:

1) Single zone or conventional system
2) Variable volume system
3) Dual duct system
4) Multizone system
5) Induction system

2.1.1. Single zone or conventional system

The conventional system is an ordinary single duct air transmission arrangement with standard air distributing outlets and includes direct control to room conditions. This system is applied within defined areas of usually constant but occasionally variable occupancies such as stores, interior office spaces and factories where precise control of temperature and humidity is not required. However this system can be arranged to satisfy very exacting requirements.

The conventional system can be further divided into three categories based on the method of control.

(a) Refrigeration capacity control

In the air conditioning system, see Fig. 2.1, a thermostat located in the space return air path is set at the desired room temperature. It controls directly the refrigeration capacity of the dehumidifier, either by on-off, step or modulation controls. The choice of the control method depends on the size and type of the refrigeration plant. The resultant temperature and humidity conditions are only relatively constant, since the refrigeration machine capacity does not always match the load. The supply air is transmitted through low velocity air ducts and distributed in the space

by standard outlets or diffusers.

This type of conventional system is used in many different applications, however, the performance is best in spaces with loads that have relatively stable characteristics and minimum ventilation requirements. For this application, equipment selected to match the load is economical to operate and being loaded most of the time, maintains the space conditions at nearly constant level.

(b) Face and bypass damper control

A variation of the preceding arrangement to improve the control of space conditions and to allow a more economical selection and utilization of the refrigeration plant is the use of an air connection between the return air and the fan intake to allow a bypass of air around the dehumidifier. This arrangement,shown in Fig. 2.2, for mixing the bypassed return air with the dehumidified air improves the control of space conditions.

The refrigeration capacity is indirectly controlled by the falling temperature of the cooling medium as the dehumidifier face dampers close and the load on the dehumidifier falls off. This has a beneficial effect on the humidity since the temperature of air leaving the humidifier tends to fall with the decreasing air bypass factor and the falling temperature of water. When the face dampers are closed, the refrigeration equipment is stopped. In marginal weather the refrigeration equipment is shut down. The face and bypass dampers are set open and closed respectively.

(c) Reheat control

The reheat system is a modification of the single zone system. Its purpose is to permit some degree of control for areas of unequal loading. Reheat implies the application of heat as a secondary process, to either preconditioned or recirculated air. A single low pressure reheat system is produced when a heating coil is inserted in the duct system. The more sophisticated systems utilize higher pressure duct designs and pressure reduction devices to permit system balancing at the reheat zone.

Air is supplied at dewpoint and drybulb temperatures low enough to balance the maximum expected cooling load in any space,see Fig. 2.3, and reheat is supplied in the branch duct to each zone,as required to match the actual sensible heat load. In the case of a single zone system, the reheater is usually located at the apparatus. Zone reheat provides excellent control under highly variable conditions and the air system is stable, but at the expense of supplying more cooling than actually required and cancelling part of it with heat.

2.1.2. Variable volume system

The variable volume system compensates for varying load by regulating the volume of air supplied through a single duct. Special zoning is not required because each space supplied by a controlled outlet is a seperate zone.

Significant advantages of the variable volume system are low initial cost and low operating costs. Since the volume of air is reduced with a reduction in load, the

refrigeration and power follows closely the actual air conditioning load of the building. All outdoor air is available during intermediate seasons for cooling.

A high or low velocity air duct system is used to move the air from the apparatus to the room terminal units. The supply air quantity varies directly with the load. Duct construction is important because in areas where there is complete absence of thermal load, duct pressure can build up to the fan discharge pressure. Consequently it is necessary to construct the ducts for that pressure. The ducts must not only withstand a variable pressure, but must also be sealed to prevent air leakages.

Fig. 2.4 shows a variable volume system,and parallels the reheat system except (i) the dehumidifier is sized for instantaneous peak loads of zones involved and (ii) individual reheaters are replaced by air volume control applied either to individual branch ducts or individual outlets.

As the air quantity delivered by a supply grille or diffuser is reduced, its throw also reduces and its drop increases. This results in cold draughty conditions near supply air devices and it restricts the minimum supply air quantity to about 50 per cent of design flow. With some volume supply devices,as the demand for the supply reduces, the cross-sectional area of the supply ppening reduces, the velocity is kept constant and the controllability of the variable volume devices can be extended to perhaps 20 per cent of the design volume.

It is apparent that there is some risk of overcooling if the sensible heat gain diminishes to a value beneath the controllable range of the variable unit. This sort of situation may be dealt with by terminal reheaters. This system is applied in multizone areas and is seldom used in zones of high load fluctuation.

2.1.3. Dual air system

This system provides the choice of heating and cooling as required by different zones through dual air streams. Air is supplied to the two air streams at different conditions (usually one hot and the other cold) and mixed by proportioning dampers either in the room upstream or in a plenum. The entire air quantity for absorbing the load is conditioned centrally and distributed by the main fan or fans. Either high- or low velocity duct distribution may be used. If desired, mixing may be performed at the apparatus with only a single duct extending to each zone.

To maintain conditions at all times, the cooling conditioner and ducts must be sized for cooling load that exists when no heating is required from the other duct and vice versa. This means that for any system where there is a wide variation in the cooling and heating requirements among the space served, each conditioner and duct will have to be sized to carry possibly 75 per cent or more of the maximum load. At peak summer load both airstreams can aarry the cooling effect, provided the equipment is arranged to furnish it.

For best performance some form of constant volume regulation should be incorporated into the system to maintain a constant flow of air. Without this, the system is difficult to control because of the wide variations in system static pressure that occur from the normal demand from lodding changes.

The following arrangements are commonly used.

(a) Dehumidification of cold stream and reheat of hot stream

One portion of the mixture of ventilation and return air is cooled and dehumidified, the remainder is heated. This arrangement is open to the serious objection of by-passing outside air directly into the room , as shown in Fig. 2.5, and is tolerable only where the proportion of the outside air is very small - say less than 10 per cent.

(b) Dehumidification of minimum outside air

By adding an extra dehumidifier in the above arrangement, as shown in Fig. 2.6, a better control of humidity can be achieved, but at the expense of higher cost.

(c) Dehumidification and cooling of all supply air with reheat for hot stream

The air from the dehumidifier and fan is split and a portion is heated to provide the hot stream, see Fig. 2.7. Control is simple but economy is poor because of the cancelling of cooling by reheat.

Apart from the physical arrangement, the above three systems differ only in the precision with which relative humidity can be controlled.

2.1.4. Multizone system

The multizone system provides a single supply duct for each zone and obtains zone control by mixing hot and cold air at the control unit in response to room or zone thermostats. For a comparable number of zones this system provides greater flexibility than the single duct and involves lower cost than the dual duct system, but it is physically limited by the number of zones that may be provided at each central unit. A typical multizone system is similar in some respects to the dual duct system but the two air streams are proportioned within the equipment instead of being mixed at each space served. The proper temperature air is provided as it leaves the equipment.

From a central station apparatus (usually built up package) the multizone system, see Fig. 2.8, distributes a single stream to each room or zone through separate ducts. The central station includes dampers that premix the proper amounts of cold and warm air for each duct and is controlled by room apparatus thermostats.

Multizone system is considered when one or more of the following conditions exist:

i) The area consists of several spaces to be individually controlled.

ii) An area with zones of different exposure and different internal loads.

iii) A large interior zone combined with a relatively small groups of exterior places.

2.1.5. Induction system

The design and operation of an induction system is based on the Air/Transmission ratio concept. The A/T ratio is the ratio of the unit primary air quantity to the total room transmissions per degree temperature difference through the external areas of the room served by the unit. The transmission per degree is the steady state heat loss for one degree temperature difference across external windows, walls and roof. The system is designed so that the primary air quantity to a given room is proportional to the transmission of that room. Using the A/T ratio concept a room with 50 per cent greater heat loss would receive 50 per cent more primary air. Thus reheat to all rooms in a zone may be scheduled in accordance with changes in outside air dry bulb temperature.

The induction system selected for a given space must be able to:

i) Supply a quantity of air to the space in a fixed proportion to the transmission per degree of the space. The outdoor air portion of the supply air must equal or exceed the ventilation required.

ii) Produce a total cooling capacity that equals or exceeds calculated room sensible heat gain.

iii) Operate at a nozzle pressure consistent with an acceptable sound level.

An induction system may either be a constant temperature or a constant volume induction system. A constant volume induction system, shown in Fig. 2.9, is particularly used for medium and small multiroom buildings, where

individual rooms as well as large spaces may be air conditioned from one central air conditioning plant. It is often applied to buildings having a large ratio of floor area to height, indicating a need for horizontal ductwork and piping. This system is particularly suited to high latent heat load applications.

A constant volume of cool conditioned air is supplied to the induction unit, see Fig. 2.10. This air designated primary air handles the requirements for cooling, dehumidification or humidification and ventilation. The primary air induces room air which is heated by the coil to provide summer tempering and winter heating.

Room temperature control is achieved by adjusting the flow of hot water or steam through the coil by a manual or a an automatic control valve. Supply air temperatures may be as much as 15° C below room temperature since room air is mixed with the primary air before the total air stream is discharged into the room. This feature makes possible smaller air quantities at lower temperature. Also this means smaller duct sizes and smaller central station apparatus.

2.2 Air-water systems

By physically separating the sensible and latent heat removing functions, through the use of separate heat exchangers, these systems have important advantages in multi-room comfort applications where relatively high sensible heat factors occur.

Temperature is controlled by throttling the source of sensible cooling (usually water), while the humidity is maintained within acceptable limits by fixing the dew point of the primary air supply. With higher sensible heat ratios, the quantity of primary air required to absorb the moisture load at a practical apparatus dewpoint is only a fraction (usually one fifth to one half) of the air required to absorb the sensible heat. Since the primary air required for dehumidification is comparable to the ventilation rate, it is logical to introduce and precondition the ventilation rate in the primary air conditioner.

Air-water systems can be divided into:

1) Primary air induction system
2) Primary air fan-coil system

2.2.1. Primary air induction system

In this system the conditioned primary air is delivered at pressures of 250 to 500 Nm^{-2} to a room induction unit, see Fig. 2.11, from the nozzles of which it issues at velocities of 20 to 30 ms^{-1} entraining secondary (room) air over the coil. The flow of water through the coil is manually or automatically throttled to balance the sensible load and maintain the room temperature. Induction ratio of three to five volumes of room air to primary air provides satisfactory room-air motion and distribution.

Chilled water from a central refrigeration plant is circulated by a primary water pump through the dehumidifier coils in the apparatus and a secondary water pump is used for the induction unit coils.

Induction units have single row high-surface-ratio water coils and cooling capacities ranging from 880 to 2,640 W under normal operating conditions with primary air quantities of 0.0142 to 0.0944 m^3s^{-1}.

2.2.2. Primary air fan-coil system

The primary air fan-coil system is in many ways similar to an induction system, the basic difference being the substitution of a small fan in place of induction nozzles to circulate room air across a water coil.

The advantage of the fan-coil system is its flexibility in design and layout. The units may be vertical or horizontal type for visible or concealed application. Though there may be a disadvantage associated with low efficiency and low powerfactor of numerous small power motors.

Fig. 2.12 shows a fan-coil system. In climates with well-defined summer and winter seasons a changeover system is designed so that chilled water is provided in summer and warm secondary water in winter.

2.3 All-water systems

All-water systems are systems using fan-coil, unit ventilator or valance-type room terminals, with unconditioned ventilator air supplied by an opening through the wall or by infilteration. Cooling and dehumidification are provided by circulating chilled water. Humidification is not practical in all-water systems unless a separate package humidifier is provided in each room.

A big advantage of an all-water system is its flexibility for adaption to many building module requirements. This system has individual room control with quick response to thermostat setting and freedom from circulation of air from other conditioned spaces.

All-water systems use fan-coil units to achieve the required air motion in the conditioned space. These fan-coil units are similar to those used in air-water systems except that drain pan is required since dehumidification takes place together with sensible cooling.

All-water systems can be classified according to the water piping system used. The water piping systems are divided into once-through and circulating types. In a once-through system water passes through the equipment only once and is discharged. In a recirculating system water flows in a repeating circuit from the heat exchanger to the refrigeration equipment and back to the heat exchanger.

The recirculating system is further classified according to water return arrangements. When two or more units are piped together, one of the following piping arrangements may be used:

1) Reverse return system
2) Direct return system
3) Reverse return headers with direct return risers system

2.3.1. Reverse return system

This system is recommended when the units have nearly the same pressure drop through them. Reverse return system is often the most economical design on new construction. The length of water circuit through the supply and return piping is the same for all units, hence this system seldom requires balancing valves, see Fig. 2.13a.

2.3.2. Direct return system

When the units have different pressure drops, it is more economical to use direct return system and balancing valves, as shown in Fig. 2.13b.

The number of balancing valves can be reduced by designing the supply and return pressure drop equal to one-fourth the sum of the following pressure drops:

(i) the loss through the supply and return runouts from the riser to the unit,
(ii) the loss through the unit itself and
(iii) the loss through the fittings and valves.

2.3.3. Reverse return headers with direct return risers system

There are installations where it is both inconvenient and economically unsound to use a complete reverse return system. This sometimes exists in a building where only top floors are to be conditioned. In this case reverse return headers are located at the top of the building and direct return risers to the units are used. Such a system is shown in Fig. 2.14.

All-water systems may use single, double or multi pipe system. In a single pipe system a single supply of water (cold or hot depending on the season) is available at each room unit. For applications where both heating and cooling loads exist in a building, a double pipe system may be designed using two pipes carrying hot and cold water with a single return pipe. However in some cases it may be more economical to design a multi pipe system in which two seperate return pipes for hot and cold water are used.

2.4 Unitary and other systems

2.4.1. Unitary system

In this system both heat and moisture removal are performed by the room unit; and outside air ventilation is supplied through an opening in the outside wall, from a supplementary ventilation system or in some cases none at all.

Unitary system consists of factory matched refrigerant cycle components for inclusion in air conditioning systems that are field designed to meet the needs of the user.

The following list of variations is indicative of the vast number of types of unitary air conditioners presently available:

1. Arrangement: single or split
2. Heat rejection: air cooled, evaporative condenser, water cooled
3. Unit exterior: decorative for in-space application, functional forequipment room and ducts, weatherproofed for outdoors

4. Placement: floor standing, wall mounted, ceiling suspended
5. Indoor air: vertical upflow, counterflow, horizontal, 90 and 180 degree turns, with fan, or for use with forces air furnace
6. Locations: indoor-- exposed with plenums or furred in ductwork, concealed in closets, attics, crawl spaces, basements, garages, utility rooms, or in equipment rooms; wall-- built in, window, transom; outdoor-- rooftop, wall mounted, or on ground

2.4.2. Panel system

This system incorporates the features of panel cooling and heating into a central system design. Metal ceiling or wall panels, through which chilled or hot water is circulated, absorb part of the sensible cooling or heating load primarily by radiation. Supplementary air systems are applied to offset the remaining sensible load as well as to serve the ventilation and dehumidification or humidification functions. Careful control of chilled water and room dew-point temperatures is required to prevent condensation on cold panel surfaces. Concentrated cooling loads, such as from business machines and high wattage lighting loads, may be absorbed directly by radiation instead of being added to the room sensible load to be removed from the air. Conditioned air may be used instead of circulating water for panel cooling prior to being discharged into the room.

2.4.3. Evaporative system

When unsaturated air is circulated over a surface of warm water, moisture will be evaporated until the air becomes saturated. For this process to occur, the latent heat of vapourization must be supplied from some source. Initially part of this heat will come from the remaining unevaporated water which will be cooled until it reaches the temperature of evaporation after which its temperature will remain unchanged. During this latter stage,if there is no other source of heat, all of the latent heat must be supplied by the air itself; that is, air will be cooled by evaporation of water into it. The minimum temperature to which air can be cooled in this fashion is the temperature of adiabatic saturation.

This inexpensive means of cooling without mechanical refrigeration finds application in hot dry areas such as semi-arid regions where it is possible to obtain some measure of relief by removing sensible heat only. There are two types of evaporative air coolers.

(a) Direct type of evaporative air cooler

This type uses primary wet surfaces which evaporate water into the fresh air supplied to the space being cooled. This may be achieved with a bank of sprays, as in an air-washer, or by use of extended surface material. The three main types of direct air cooler are:

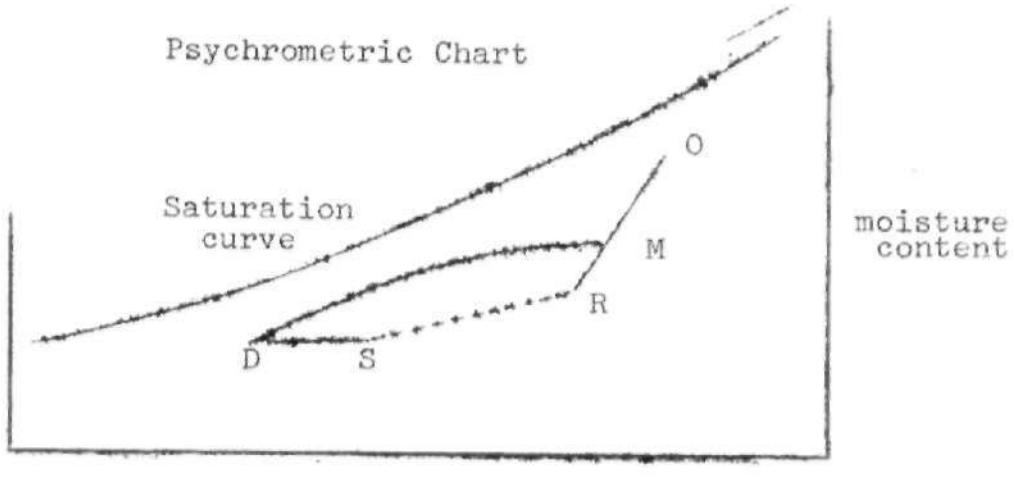
(i) Drip (or desert) type cooler - This consists of of a fan unit with evaporative pads over which water is passed by a small circulating pump

(ii) Spray type cooler - This type contains evaporative pads and a fine water spray which is thrown into the air and onto the pads by water slinger,aa centrifugal vapourizer or by spray nozzles.

(iii) Rotary type cooler - There is a device which conti - ously wets and washes the evaporative pad by rotating it through a water bath and presenting it to the air stream.

(b) Indirect type of evaporative air cooler

The indirect type ofcooler uses the evaporative principle to cool water in a device which is similar to a cooling tower, the resulting chilled water being passed through a coil to cool the air which is supplied to the occupied space. By arranging a precooling coil at the air inlet of cooling tower, and diverting part of the cooled water to cool its own precooling coil, it is possible to achieve water temperatures even lower than the origional wet bulb temperature.



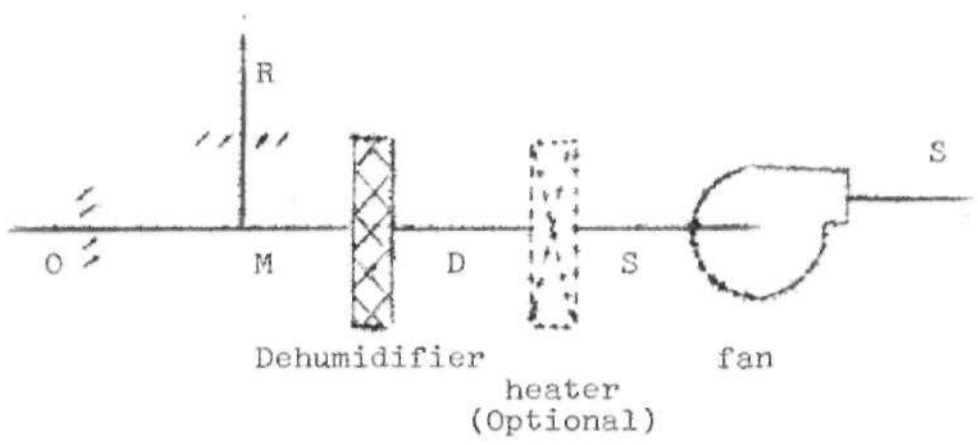


Fig.2.1 Single zone system
(Refrigeration capacity control)

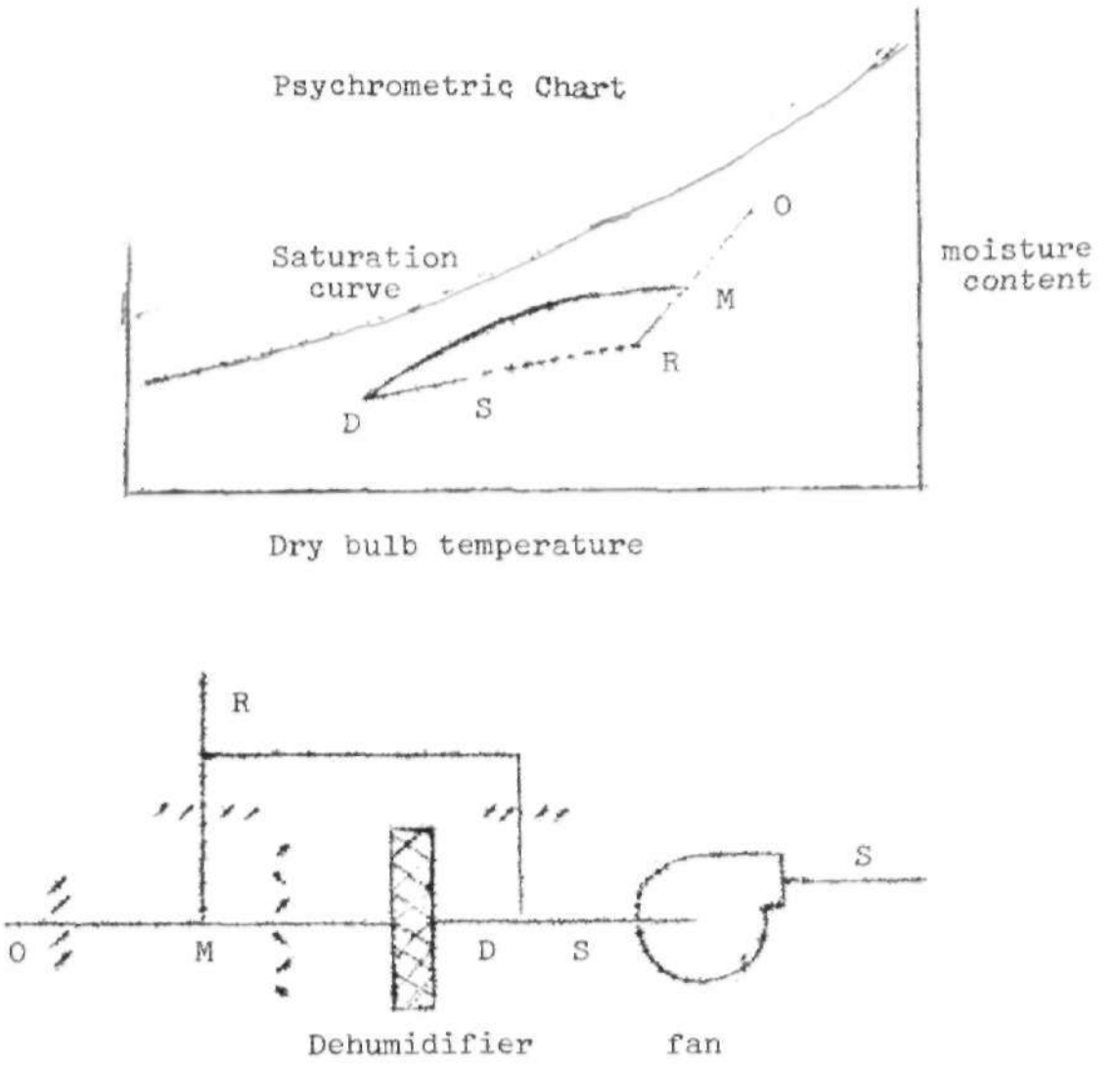
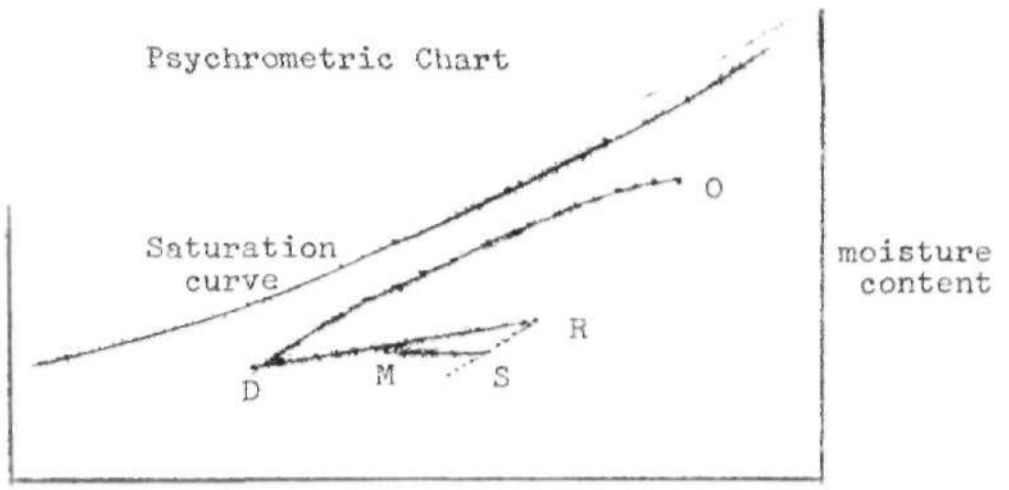


Fig. 2.2 Single zone system
(Face and bypass damper control)



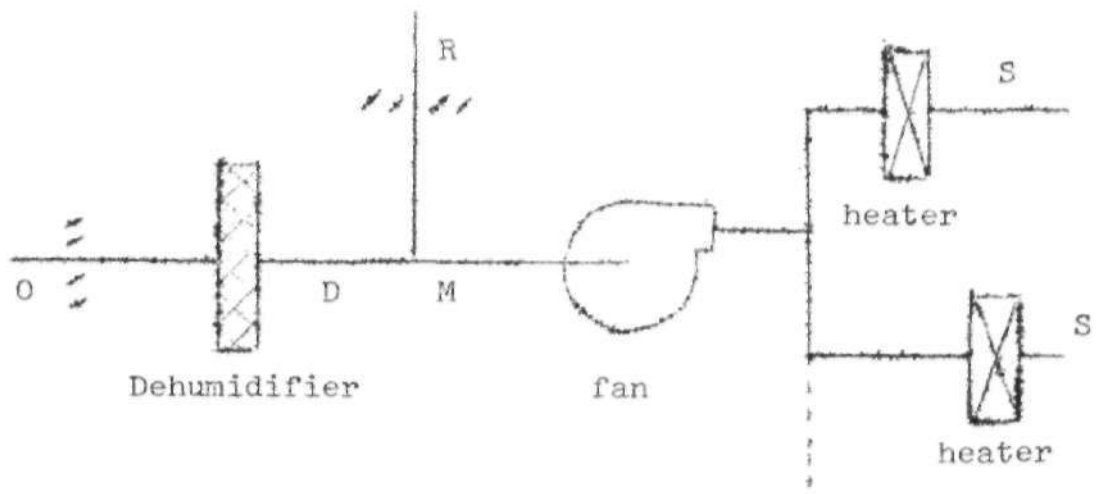


Fig. 2.3 Single zone system
(Reheat control)

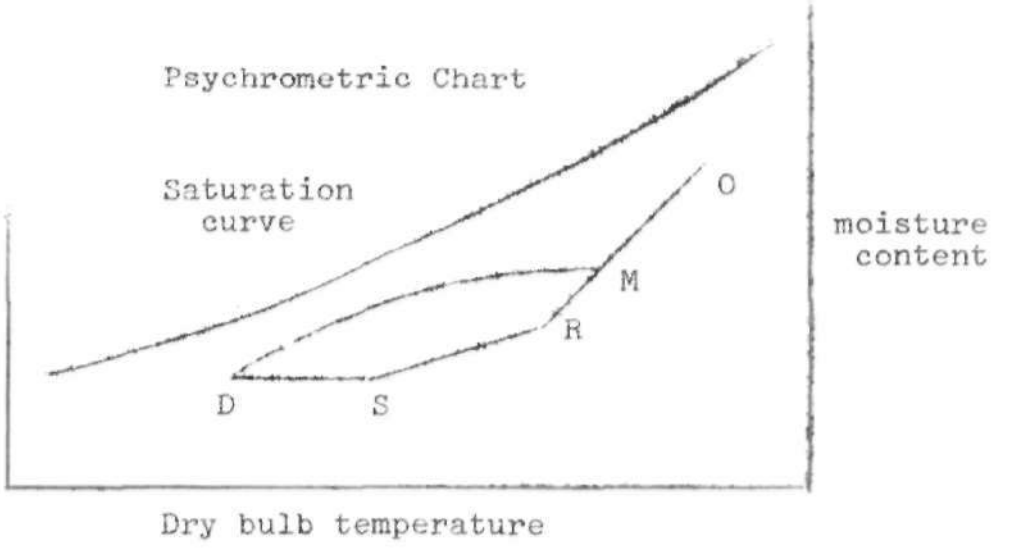


Fig. 2.4 Variable volume system

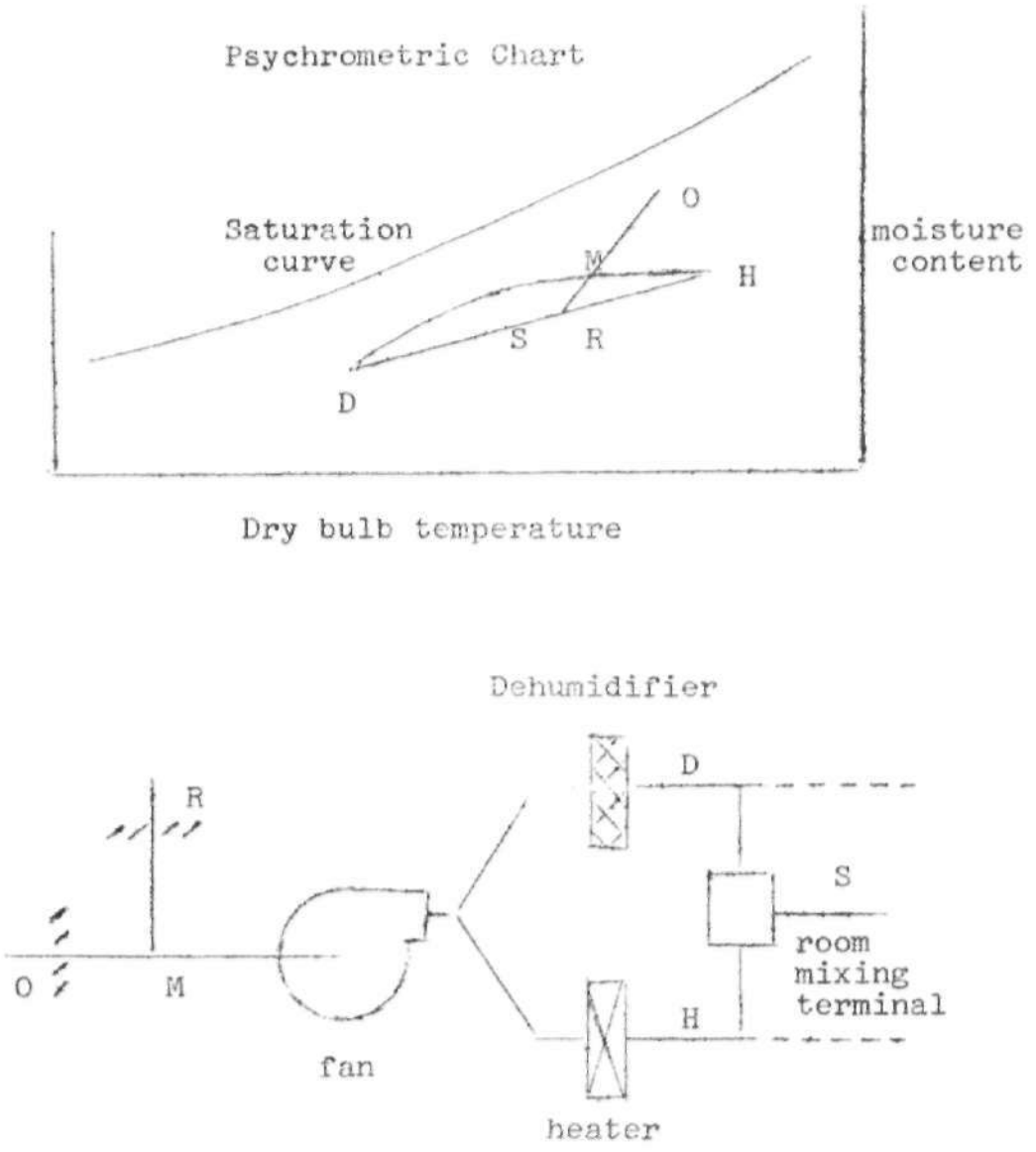
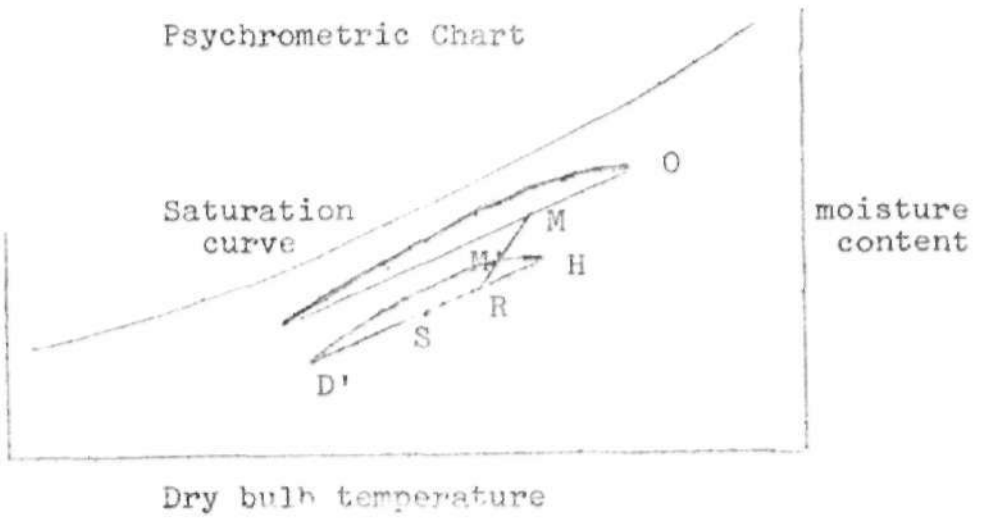


Fig. 2.5 Dual air system
(Dehumidification of cold stream)



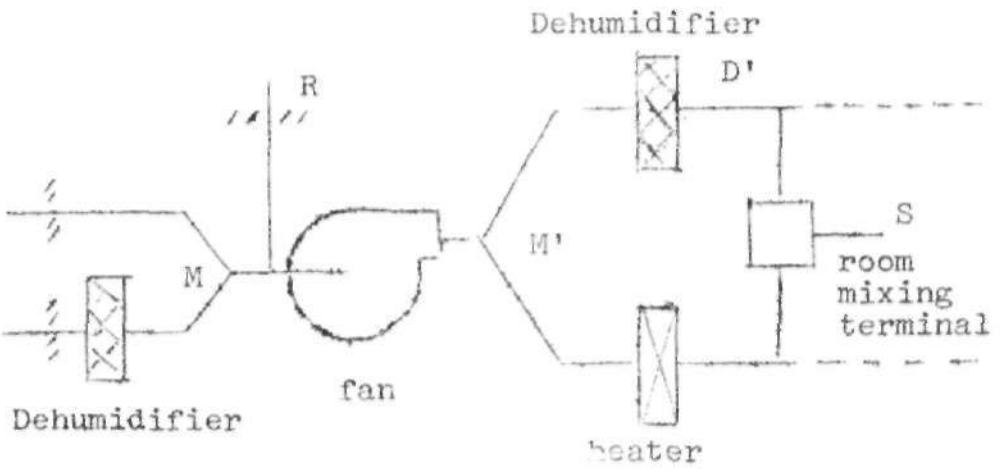


Fig. 2.6 Dual air system
(Dehumidification of min. outside air)

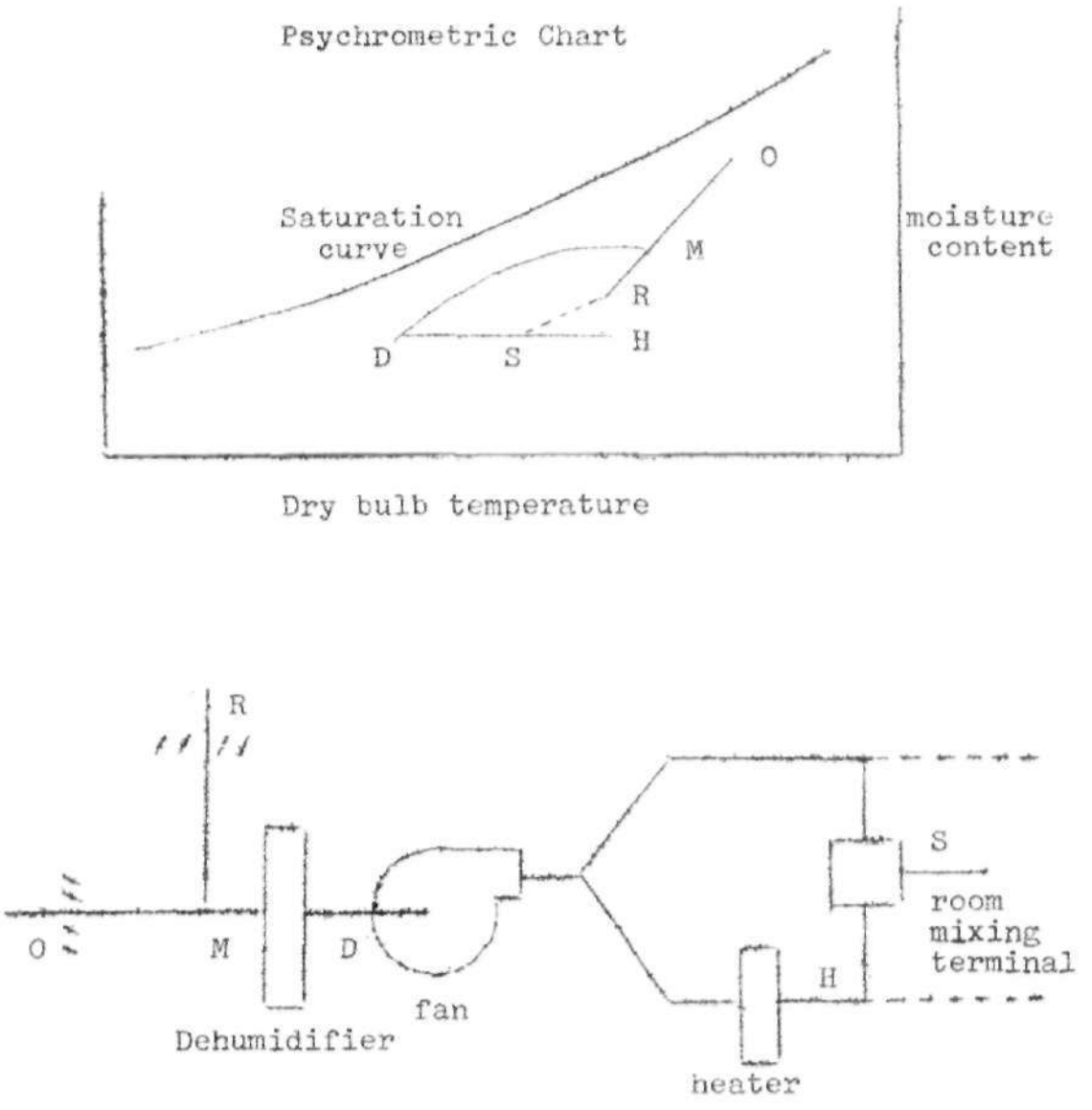
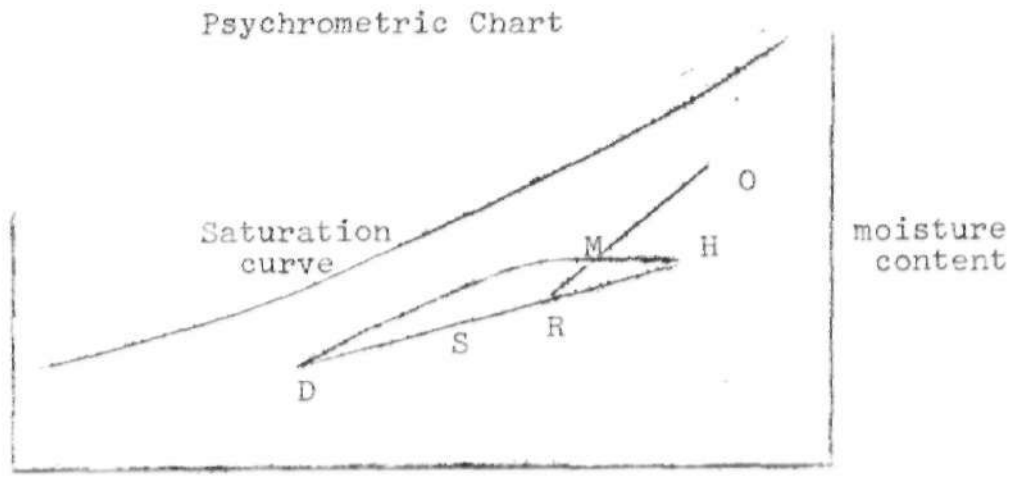


Fig. 2.7 Dual airsystem
(Dehumidification of all air)



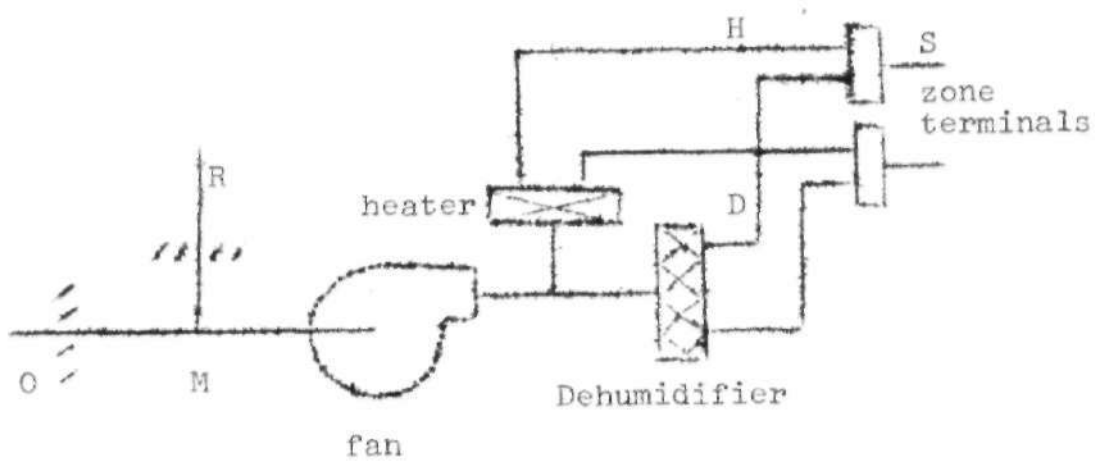


Fig. 2.8 Multizone system

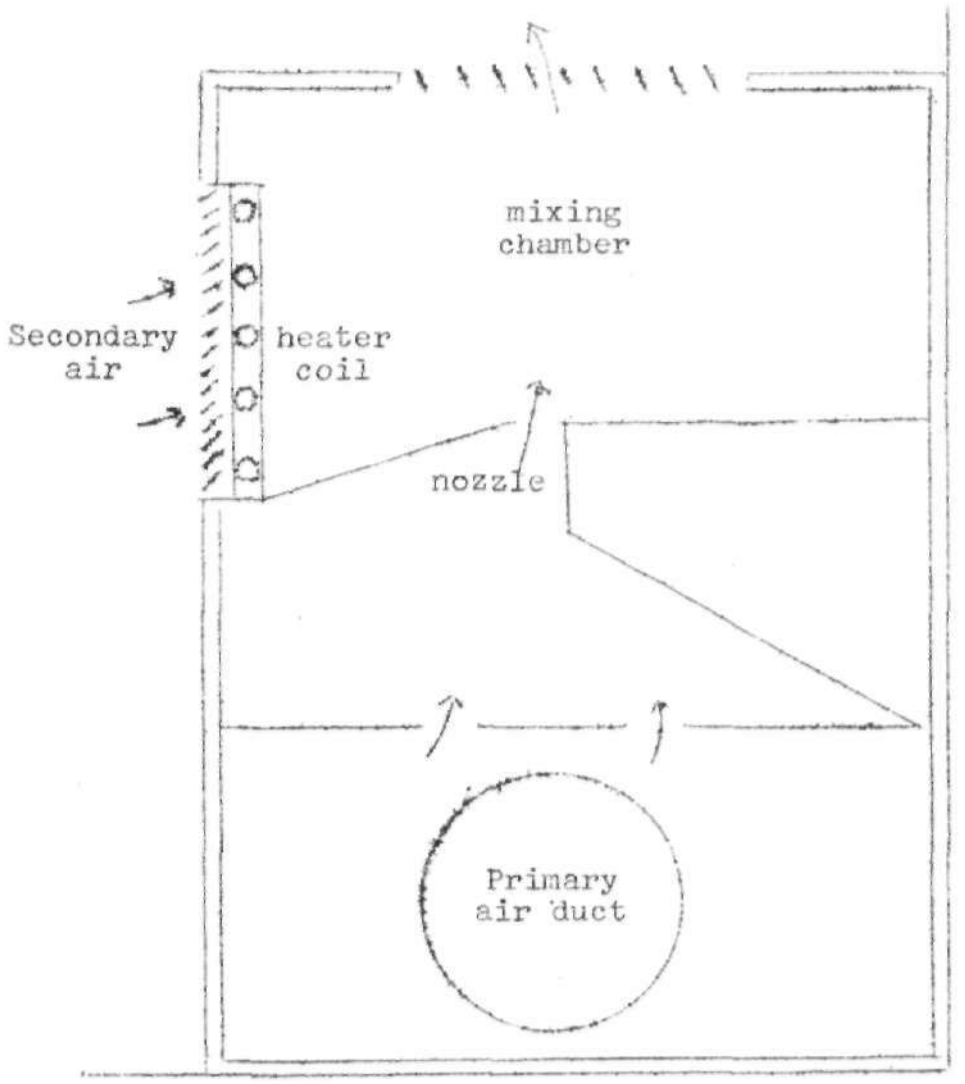
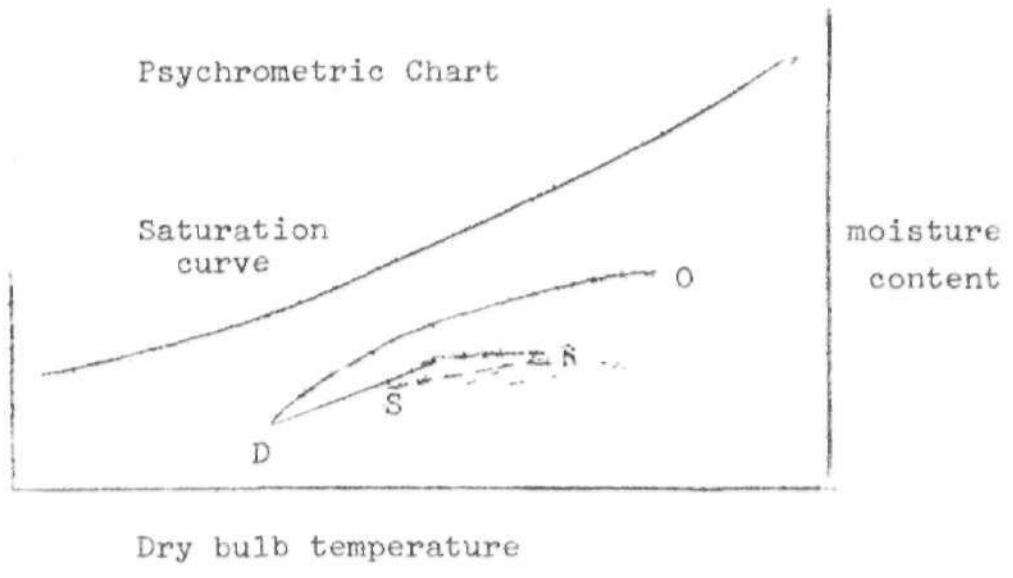


Fig. 2.10 Induction unit



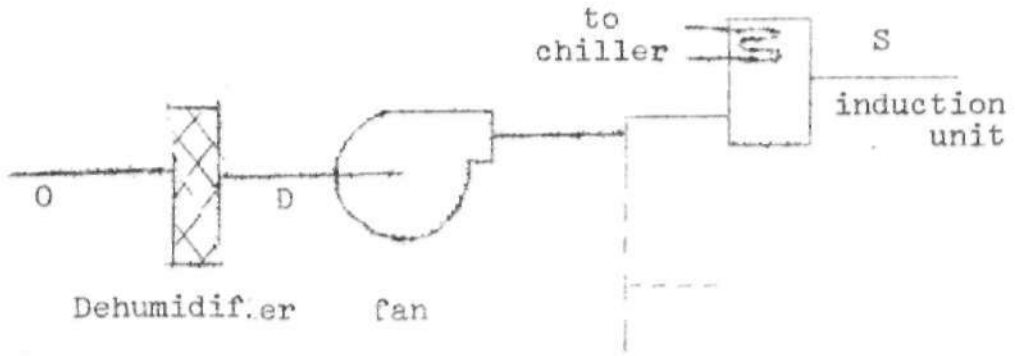
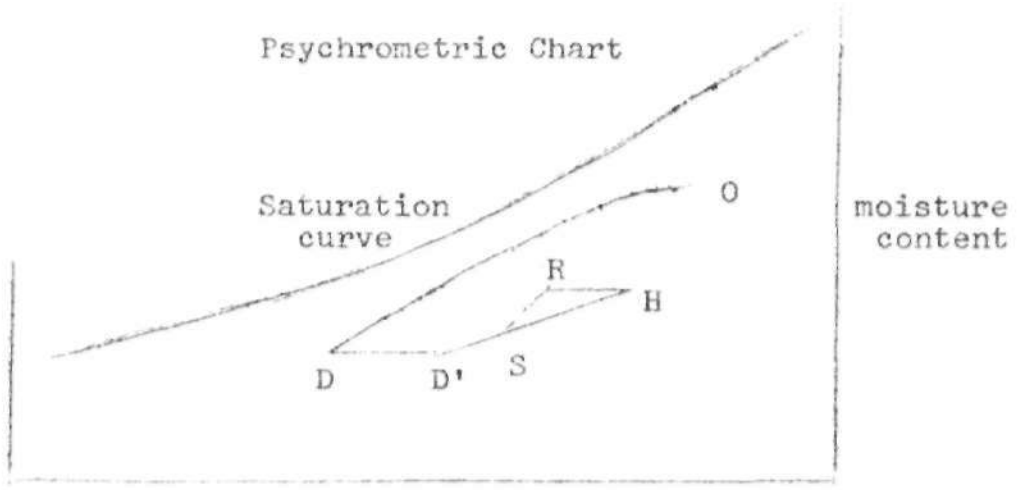


Fig. 2.11 Primary air induction system



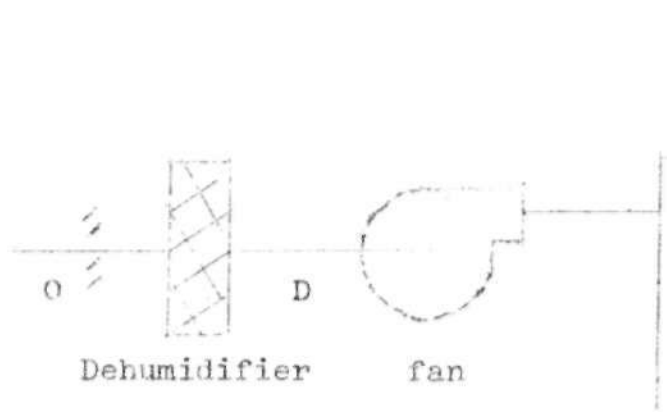
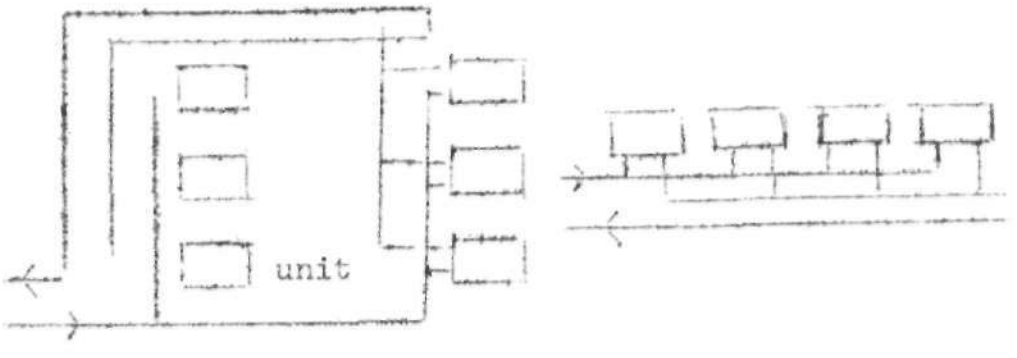


Fig. 2.12 Fan-coil system

a) Reverse return system

vertical layout horizontal layout



b) Direct return system

vertical layout horizontal layout

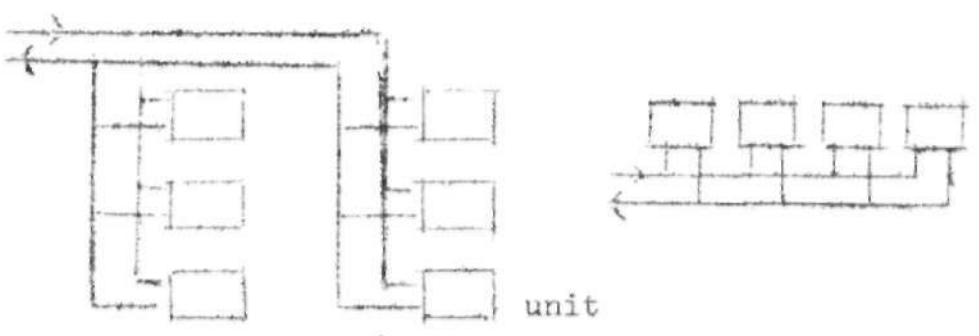


Fig. 2.13 Watwr piping systems

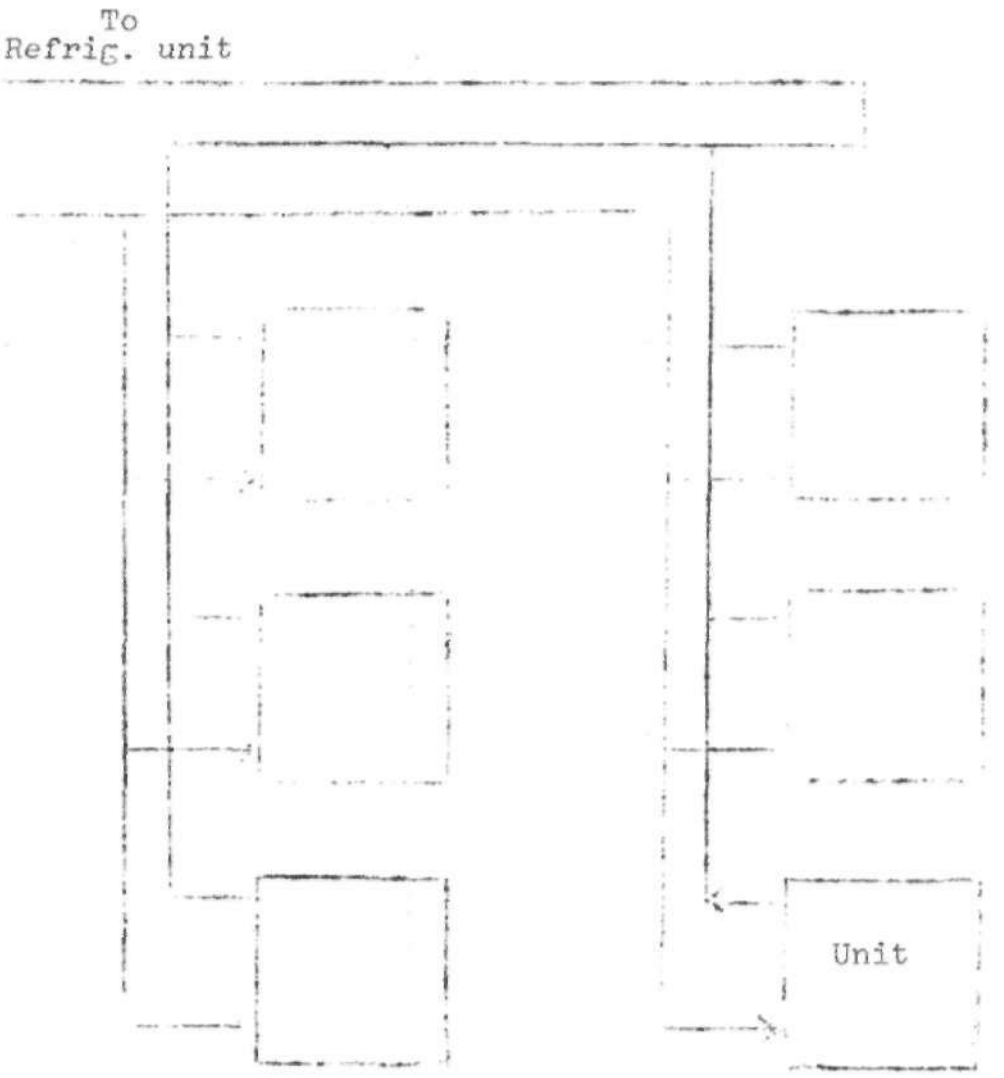


Fig. 2.14 Reverse return headers with direct return risers - water piping system.

CHAPTER 3

SYSTEM SELECTION AND LOCATION

3.1 System selection

It can be seen from the previous chapter that the choice of HVAC system is wide and that the types and combinations of equipment are numerous. The selection of air conditioning system depends on many factors; the most important among them being the shape, function and material structure of the building, the initial and operating costs.

3.1.1. Mechanical Engineering building

The Mechanical Engineering building is 13 m wide and about 60 m long and consists of four floors:

1) Ground floor: This floor contains mainly laboratories and is not to be air conditioned
2) First floor: This floor has offices and a conference room. It is already air conditioned by two seperate unitary air conditioners. The larger space being air conditioned by a split condensing unit system
3) Second floor: This floor has a library, a colloquium room, three research rooms and a photographic dark room and a Part III lecture room, having access from a 1.80 m wide corridor.
4) Third floor: This top floor contains Part II lecture room, a 4.5 m by 4.5 m room, and Part I Drawing room.

The Drawing room, 40 m long, has two sliding panels which can be used to divide the room into three equal parts

Fig. 3.1 shows the second and third floors of the Mechanical Engineering building. The rooms have been numbered from one to nine starting from the top floor.

3.1.2. Economic considerations

The importance of a complete economic study should not be overlooked by the designer and the purchaser of air conditioning equipment. Too often minimum first cost is given undue weight, without proper evaluation of operating economics. In industrial applications, the returns on the cost of installation and running an air conditioning system can be assesed quantitatively and the economic justification clearly demonstrated. The value of air conditioning is less easy to asses when a building is air conditioned purely for comfort of the occupants, since there is a lack of data by means of which a monetary value can be placed on human comfort and well-being.

The cost of an air conditioning equipment can be divided into:

a) Installation and owning cost

b) The owning element of the cost is influenced by the price of the equipment, materials, labour and services for an installation. These components must be weighed carefully before arriving at the selection of a system

b) Operating expences

Operating costs are influenced by the consumption of energy. There is also the item of maintenance consisting of operating personnel and upkeep of equipment together with supplies of oil, filters and other materials to keep equipment working at conditions approaching installation conditions except for normal duty wear.

The Mechanical Engineering building space may be air conditioned using unitary air conditioners. One or more may be required to condition one room in order to obtain proper air distribution.

The cost of this type of conditioner may be analysed if the following units are used:

Room 1	four 1.12 kW units
2	six 1.12 kW units
3,4	two 0.75 kW units in each
5,6,7,8	one 0.75 kW unit in each
9	four 0.75 kW units

the cost would be:

10 1.12 kW units at ₦ 675 each :	₦ 6,750
12 0.75 kW units at ₦ 575 each :	₦ 6,900
Installation and labour charges @ ₦70 each :	₦ 1,540
Total first costs :	₦ 15,190

In a central system the refrigeration equipment is located in one place from where the cooling effect is distributed by either water piping or air ducts.

An all-water or air-water system could be used if a suitable place can be found for the chiller. In the case when the whole building is to be conditioned a good choice would be an all-water or air-water system with the water chiller on the ground floor. The choice between an all-water and air-watersystem will depend upon the cost of air handling unit together with air ducts compared to the cost of larger piping andlarger room units required for dehumidification as well as cooling.

An all-air system can be chosen for the upper two floors of the building. A system with low first costs is the single zone system andalso the most widely available. The possibility ofusing a single zone system has to be ascertained from the cooling loads for each room at various times of the day. Although it can be reasonably assumed that the rooms will have same proportion of cooling load since most of them have south exposure heat gains.

The cost of this system may be analysed as follows:

Roof mounted air cooled condensing unit (12.28 kW)	₦ 8,630
Air handling unit (1.49 kW)	₦ 3,900
Ducts and outlets	₦ 6,000
Installation, labour charges and building improvements	₦ 2,500
	₦ 21,030

The estimated total number of hours of operation per year are worked out on the following considerations:

i) The equipment works 12 hours a day.
ii) It works on an average of 25 days in a month for eight months from March till October (both inclusive)

Thus the total number of hours of operation is

12 x 25 x 8 = 2400 hours

The consumption of electricity would be for

(i) Unitary system

2400 x (12 x .75 + 10 x 1.12) kWH = 48,480 kWH

and (ii) Central system

2400 x (12.28 + 1.49) = 33,048 kWH

The cost of electricity can be calculated from electricity tariffs.[11] **

(i) Unitary System (Single phase supply)

@ 6k per kWH the cost is 48480 x .06 naira = ₦2908.80

(ii) Central System (Three phase supply)

@ 6.5k per kWH the cost is 33048 x .065 naira
= ₦ 2148.12

Thus a savingof over ₦ 750 per year can be realized in energy costs by selecting a central air conditioning system. Therefore a central air conditioning system incorporating an air cooled condensing unit with an air handling unit is designed for the Mechanical Engineering Department building.

** Supernumerals will be used to indicate the serial numbers of the works consulted and listed in REFERENCES at the end of the thesis.

3.2 Location of the airconditioning system

The central air conditioning system consists of an air cooled condensing unit ana an air handler.

The air handling unit is located in the 4.5 m x 4.5 m room on the third floor of the building. The air ducts to rooms (1) and (2) would be taken through the adjoining walls, and to the second floor along the outside of the south wall of this 4.5 m x 4.5 m room.

The condensing unit is located on the roof above the air handler and connections made through the roof.

The systemlayout is shown in Fig. 3.2

The plan of the second and the third floors of the Mechanical Engineering Department building is shown in Drawing No. 1

An isometric view of thecondensing unit; the equipment with the adjoining ducts is shown in Drawing No.2

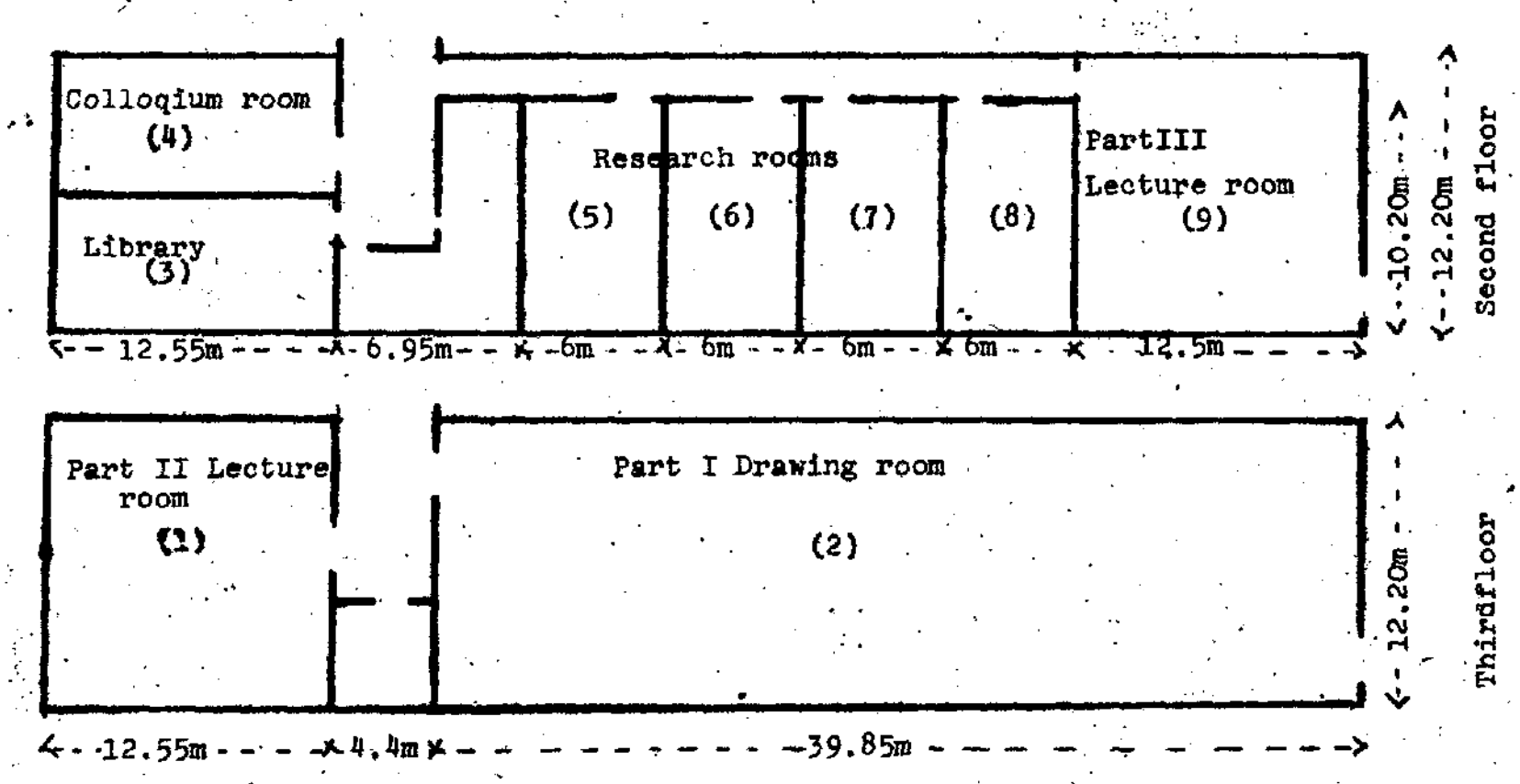


Fig. 3.1 Second and Third floors of Mechanical Engineering building, A.B.U.

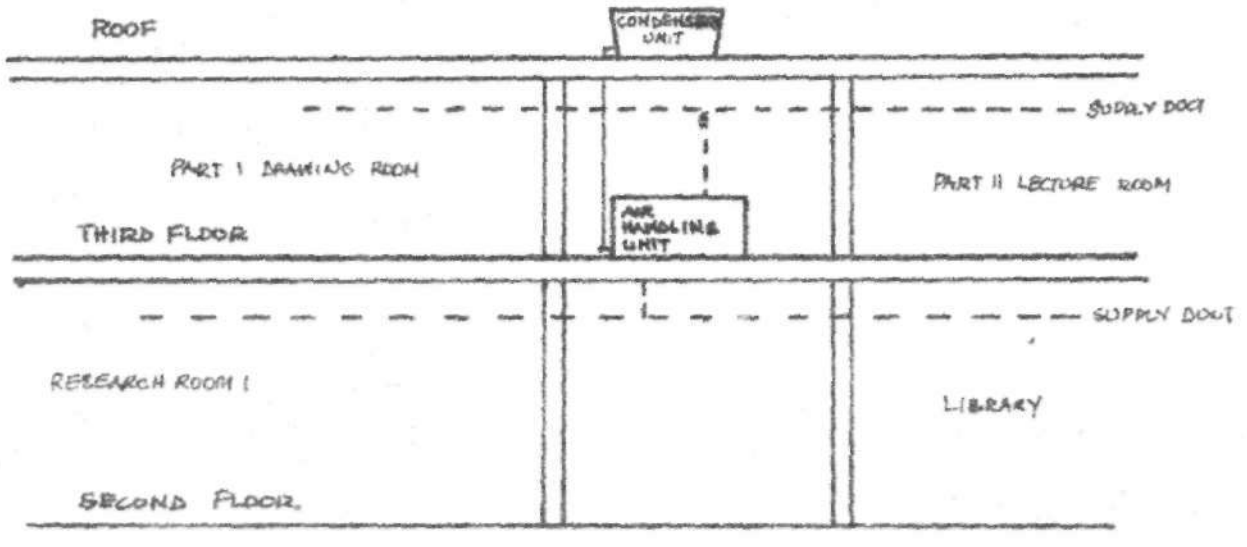


FIG 3.2 SYSTEM LAYOUT

CHAPTER 4

DESIGN DATA

4.1 Outdoor design conditions

The outdoor temperature which should be assumed for the purpose of calculating the cooling load need not be the highest ever recorded; as such temperatures occur infrequently and only for periods of a few hours duration in a year. Any rational selection of weather data should be aimed at de describing the weather on a typical warm day. The outdoor design conditions used are therefore averages of the monthly readings.

These outdoor design conditions are the simultaneously occuring dry bulb and wetbulb temperatures and moisture content, which can be expected to be exceeded a few times a year for short periods. The dry bulb is exceeded more frequently than the wet bulb temperature, and usually when the wet bulb is lower thandesign.

The Institute of Heating and Ventilating Engineers Guide gives the outdoor design conditions for Kaduna and Kano, from which design conditions for Samaru may be obtained by approprate interpolation.

Recommended outdoor conditions are:

Kaduna ($10^{o}35'$N $07^{o}26'$E; 644m above sea level)
Month of May : 36.1°C DB, 26.1°C WB, Range 10.6°C.

Kano (12°02'N 08°32'E; 467m above sea level)
Month of April: 41.7°C DB, 21.7°C WB, Range 14.4°C.
Month of June : 38.3°C DB, 26.1°C WB, Range 11.7°C.

Assuming 1°C drop forevery 110m increase in height, the outdoor conditions for Samaru are obtained as follows:

Samaru (11°10'N 07°38'E; 686m above sea level)
April: 36.3°C DB, 19.7°C WB May: 35.7°C DB, 25.7°C WB
June : 36.3°C DB, 24.1°C WB

Hence 36.0°C DB, 26.0°C WB is used for the outdoor design conditions.

When the design maximum summer temperatures are of relatively shortduration and occur in mid afternoon, it is permissible to take credit for lower outside design temperatures when the peak load occurs at other hours.

Based on a mean daily range of 8.3°C for Samaru and on the assumption of a constant dew point throughout the day, Table 4.1 shows modified outdoor desing temperatures.

Fig.4.1 shows the outdoor design conditions on a psychrometric chart together with average monthly temperatures for Samaru from dataobtained at Samaru Meteorological Station.

4.2 Indoor design conditions

The closest scientific measure of comfort is the effective temperature chart. Since temperature, humidity and air motion all influence the rate and mechanism of body heat loss they can be considered interrelated physiologically. It follows that a given combination of temperature, humidity and air motion will produce the same sensation of warmth or coolness as some other temperature with a compensating humidity and air motion. By definition,

effective temperature, shown in Fig. 4.2, is empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity and air movement.

Table 4.2 gives recommended indoor conditions which are suitable for a range of outdoor conditions from 46.0°C DB 27.8°C WB to 32.2°C DB 26.7°C WB.

4.3 Design heat transmission coefficients

In order to be able to evaluate the flow of heat per unit time across buildings, it is necessary that the overall coefficient of heat transmission be obtained. This is the time rate of heat flow per unit area for a unit temperature rise.

For a composite wall or structure of many homogeneous materials of thermal conductivities $k_1, k_2, k_3, \ldots, k_n$., the heat transmission coefficient is given by:

$$1/U = 1/f_i + \sum_{i=1}^{n} k_i/x_i + 1/f_o \qquad (4\text{-}1)$$

Where U = the overall coefficient of heat transmission
f_i : the surface coefficient of wall from inside
x_i : the thickness of ith homogeneous material
k_i : the thermal conductivity of ith material
f_o : the surface coefficient of wall from outside

4.3.1. Design heat transmission coefficient for walls

(a) Inner walls

Construction	Resistance m^2K/W
1. Inside surface air (still)	0.38
2. Lt. Wt. agg. plaster 16mm	0.22
3. Lt. Wt. agg. cement block 203mm	1.12
4. Lt. Wt. agg. plaster 16mm	0.22
5. Inside surface air(still)	0.38
	2.32

U = 0.43 W/m^2K

(b) Outer walls

Construction	Resistance m^2K/W
1. Inside surface air (still)	0.38
2. Lt. Wt. agg. plaster 16mm	0.22
3. Lt. Wt. agg. cement block 406mm	2.23
4. Lt. Wt. agg. plaster 16mm	0.22
5. Outside surface air (6.67m/s wind)	0.09
	3.14

$U = 0.32\ W/m^2K$

4.3.2. Design heat transmission coefficient for roof

Construction	Resistance m^2K/W
1. Outside surface air (3.33m/s wind)	0.14
2. Asbestus cement shingles	0.12
3. Air space 200mm	0.55
4. Asphalt roll roofing	0.08
5. Concrete slab (gravel agg.) 203mm	0.36
6. Lt. Wt. agg. plaster 12mm	0.17
7. Acoustical tile 12mm	0.66
8. Inside surface air (still)	0.38
	2.46

$U = 0.41\ W/m^2K$

4.3.3. Design heat transmission coefficient for doors

A value of $U = 0.93\ W/m^2K$ is used for a plywood door of thickness 40mm.

4.3.4. Design heat transmission coefficient for glass

Heat can be transmitted through glass by conduction and radiation. In the case of conduction a value of $U = 2.02\ W/m^2K$ is used.

Solar radiation heat gain is calculated using overall glass factors. Overall glass factors for outdoor wind velocity of 2.25 m/s and 30° angle of incidence are given for different types of glass, (fully shaded).

Type of glass		Glass factor
Ordinary (clear)		1.00
Stained :	Dark green	0.32
	Greyed green	0.46

The outside shading slats which are at an angle of 17° to the horizontal introduce an additional factor of 0.22, hence,

for main windows overall glass factor is

$$GF = .22 \times (.53 \times 1.00 + .47 \times .32) = .16$$

for louvre windows overall glass factor is

$$GF = .22 \times .46 = .10$$

Table 4.1 Design conditionsat different hours of the day[4]

Hours of the day	Correction DB	WB °C	Modified design conditions DB	WB °C
8	-7.8	-2.2	28.2	23.8
10	-5.6	-1.7	30.4	24.3
12	-2.8	-0.6	33.2	25.4
14	-0.6	0.0	35.4	26.0
16	-0.6	0.0	35.4	26.0

Table 4.2 Recommended indoor conditions for living and office spaces in tropical climates in summer[4]

Occupancy	Effective Temp.°C	Dry bulb Temp.°C	Relative Humidity %
Continuous	21.7 opt.	24.4	52
	23.1 max.	25.6	60
		26.7	45
Transient	23.9	25.6	70
		27.2	53

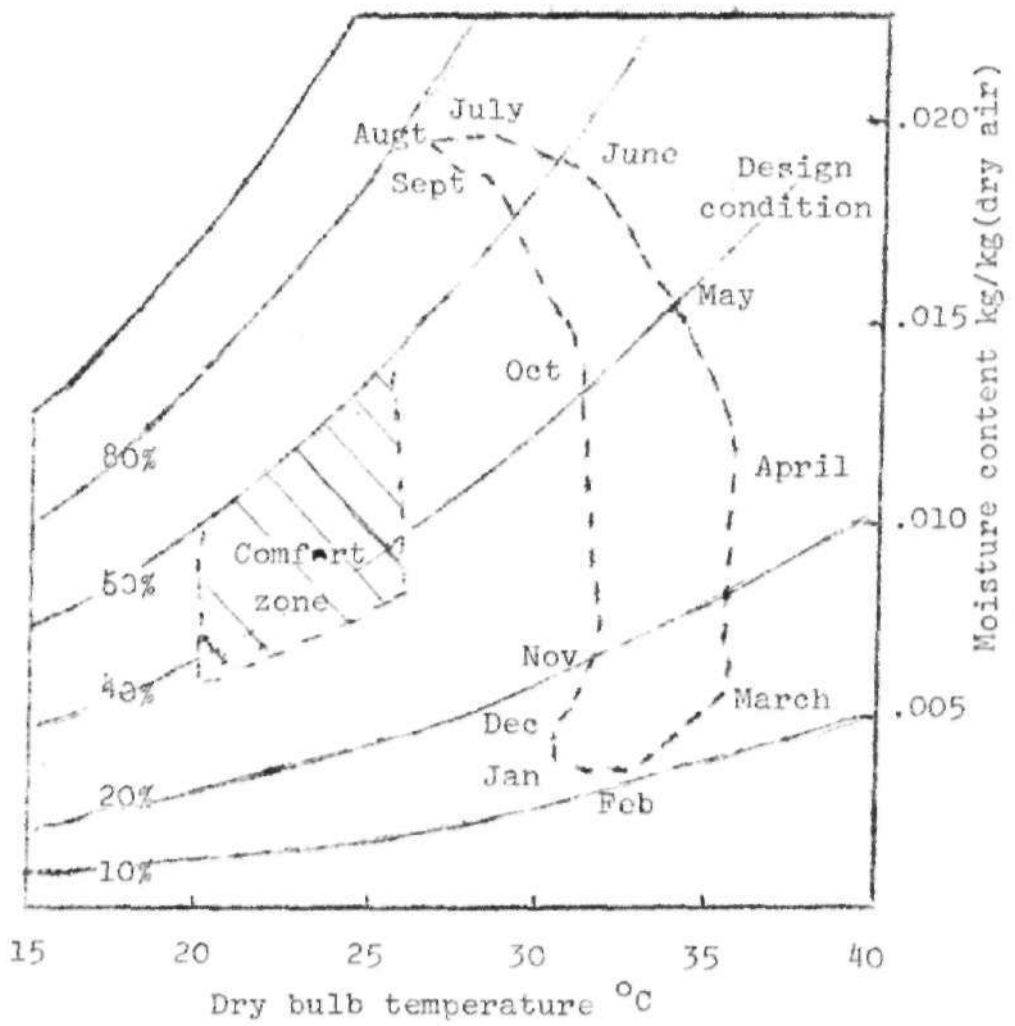


Fig. 4.1 Outdoor design conditions and monthly average temperatures for Samaru.

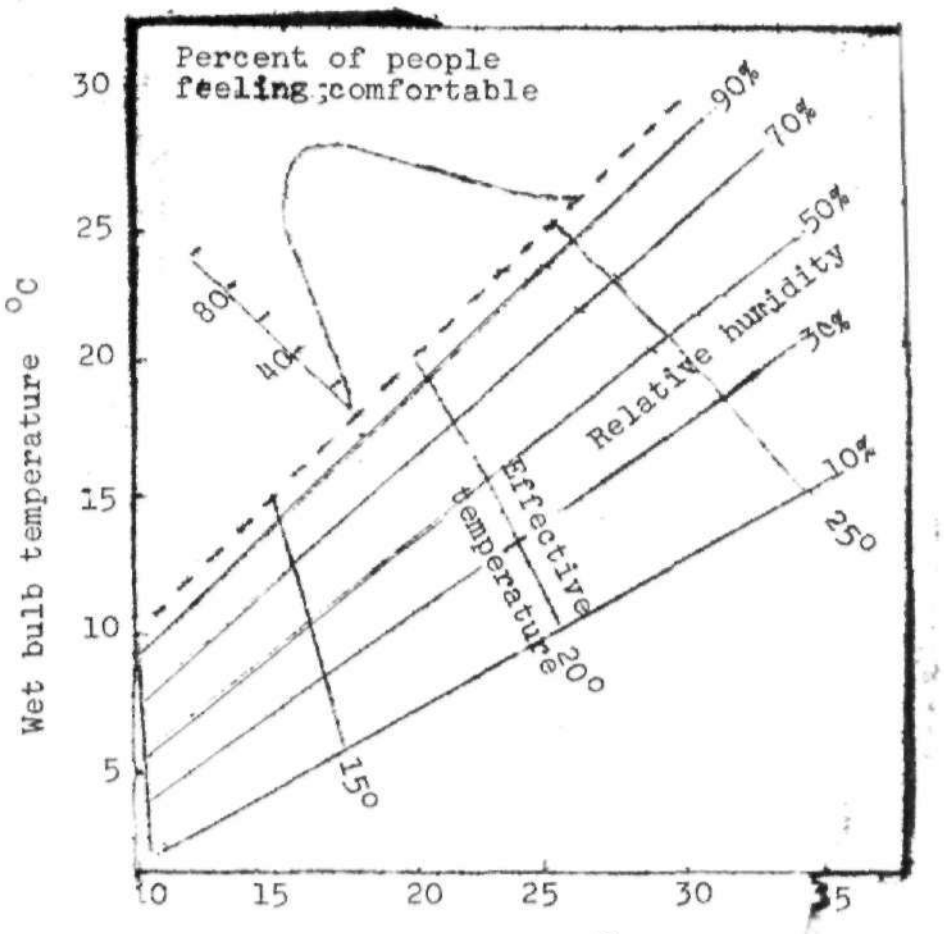


Fig. 4.2 A. S. H. A. E. Comfort Chart[4]

CHAPTER 5

HEAT GAIN FROM SOLAR AND OTHER SOURCES

5.1 Direct and diffuse radiation

The measurement of direct, diffuse and total radiation is essential to research in solar energy and in solar effects on HVAC systems.

The first reliable pyrheliometer was developed by K. Angstrom of Sweden. This instrument measures the electrical power required to heat a shaded metal strip to the same temperature as the blackened metal strip heated by the sun's rays. The response is given in the Angstrom Scale of solar radiation. In 1913 an instrument using water to cool a well insulated cylindrical black body chamber was developed by C. G. Abbot and others of the Smithsonian Institution. This led to a new scale called the Smithsonian Scale of 1913. The International Scale of 1956 is defined as the Angstrom Scale plus 1.5 percent of Smithsonian Scale minus 2 percent.

The total intensity of radiation at the upper limit of the atmosphere is 1395 W/m^2 over a spectrum stretching from wavelengths of 0.29 to 4.75 microns. Only about 1025 W/m^2 reaches the surface of the earth when the sun is vertically overhead in a cloudless sky. Of this figure about 946 W/m^2 is from radiation received directly fron the sun and the remainder from indirect solar radiation received from the vault of the sky.

5.1.1. Solar heat gain through glass

Ordinary glass absorbs a small portion of the solar heat (5 to 6 percent) and reflects and transmits the rest. The total solar heat gain to the conditioned space consists of the transmitted heat plus about 40 percent of the heat that is absorbed in the glass. Shading devices such as awnings, exterior sun screens and overhangings will reduce the total solar gain by approximately 80 percent of that for sunlit exposures.

5.1.2. Solar transmission through walls and roof

The transmission of solar heat through opaque building materials (this applies to conduction of heat through shaded surfaces due to temperature difference alone) is complicated because equilibrium is never reached. Since the transmission is far from being instantaneous there may be a lag of 1 to 24 or more hours between the incident and transmitted peaks because of the thermal resistance and the thermal heat capacity of the material. In addition the flywheel effect takes the tops off the peaks and the bottoms of the valleys, so that the difference between the maximum and minimum values of energy transmitted is usually less than those incident.

One approach to this problem is by the "Transfer function method" which will be briefly discussed here. A more satisfactory approach is to express the heat transmission by an equivalent temperature differential, whiah when multiplied

by the normal overall coefficient and the area, will give the correct final result for that material and sun orientation.

(a) Transfer function method

A transfer function is a set of coefficients that relate an output function at some specific time to the value of one or more deriving functions at that time and to previous values of both input and output functions. For example, the heat transfer from inside surface of a wall at a particular time is related through appropriate coefficients to the sol-air temperature at the same time and previous times, to the space temperature and to the heat transfer rate at previous times. This may be written as:

$$q_{i,\theta} = A\left\{ \sum_{n=0} b_n (t_{e,\theta-n}) - \sum_{n=1} d_n \left(\frac{q_{i,\theta-n}}{A} \right) - t_i \sum_{n=0} c_n \right] \qquad (5\text{-}1)$$

where b_n, d_n and c_n are the transfer function coefficients that depend on the construction of the wall or roof section.

The cooling load for a space at a particular time may be related through the use of appropriate coefficients to the heat gain at that and previous times and to the cooling load at the previous times. This is written as:

$$q_{c,\theta} = \sum_{i=1} (v_o q_{i,\theta} + v_1 q_{i,\theta-\Delta} + v_2 q_{i,\theta-2\Delta} + \cdots - w_1 q_{c,\theta-\Delta} - w_2 q_{c,\theta-2\Delta} - w_3 q_{c,\theta-3\Delta} \cdots) \qquad (5\text{-}2)$$

Calculation of heat transfer through a wall or roof section with a variable temperature in the outdoor environment and with a variable solar radiation input on the outside surface requires the heat conduction equation with nonlinear time dependent boundary conditions. In addition the wall is a complex assembly of materials and may be two dimensional. However for a single homogeneous slab the governing differential equation is:

$$\frac{\partial t}{\partial \theta} = \frac{k}{pc} \frac{\partial^2 t}{\partial x^2} \tag{5-3}$$

At $x = 0$, the outside surface, the boundary condition is:

$$-k_w \left\{\frac{\partial t}{\partial x}\right\}_{x=0} = h_o(t_o(\theta) - t_{wo}) + \dot{q}_r(\theta) \tag{5-4}$$

where $\dot{q}_r$ is the net solar radiation heat transfer to the wall at a particular time. Both t_o and $\dot{q}_r$ are functions of time. At $x = L$, the inside surface, the boundary condition is:

$$-k_w \left\{\frac{\partial t}{\partial x}\right\}_{x=L} = h_i(t_{wi} - t_i) \tag{5-5}$$

The initial condition for eq. (5-3) is some specified temperature distribution in the wall at a time equal to zero. The nonlinear time dependent boundary condition at the outside surface, eq. (5-4), is the primary obstacle in obtaining a solution to eq. (5-3).

The transfer function coefficients may be obtained for various types of walls and roofs in ASHRAE Handbook of Fundamentals (1972).

(b) Total Equivalent Temperature Difference method

The TETD is a function of time, geographical location, orientation of the suface and the composition of the wall or roof. It is assumed that wall or roof sections of similar mass and with similar overall heat transfer coefficients will have the same heat transfer rates. Tables of TETD have been prepared for certain specified conditions.[4] Obviously, if all variables were used a very large amount of material would result. Therefore a representative set of conditions have been selected.

For example, TETD for walls and roofs are based on 35°C dry bulb outdoor temperature, constant 26°C dry bulb room temperature, 11°C daily range, July 23, 40 degrees north latitude and 12 hours operation of cooling equipment. The time, surface orientation and mass of the wall or roof are allowed to vary in the tables.

For other design conditions appropriate modification to TETD are required. Correction for other latitudes is given by:

$$\Delta t_e = \Delta t_{es} + \frac{R_s}{R_m} (\Delta t_{em} - \Delta t_{es}) \qquad (5\text{-}6)$$

For light and medium coloured walls, TETD for dark coloured Walls is modified as:

$$\Delta t_e = 0.55 \Delta t_{em} + 0.45 \Delta t_{es} \qquad (5\text{-}7)$$

$$\text{and } \Delta t_e = 0.78 \Delta t_{em} + 0.22 \Delta t_{es} \qquad (5\text{-}8)$$

The eqs. (5-7) and (5-8) may be combined with eq. (5-6) to give TETD for light coloured wall as:

$$\Delta t_e = 0.55 \frac{R_s}{R_m} \Delta t_{em} + (1-0.55 \frac{R_s}{R_m}) \Delta t_{es} \qquad (5-9)$$

and for medium coloured wall as:

$$\Delta t_e = 0.78 \frac{R_s}{R_m} \Delta t_{em} + (1-0.78 \frac{R_s}{R_m}) \Delta t_{es} \qquad (5-10)$$

for any latitude,

where Δt_e is the TETD for month and time of the day desired

Δt_{es} is the TETD for the same wall or roof in shade at desired time of the day corrected if necessary for design conditions

Δt_{em} is the TETD for wall or roof exposed to the sun for desired time of the day, corrected if necessary for design conditions.

R_s is the maximum solar heat gain in W/m^2 through glass for wall facing or horizontal roof for month and latitude desired

R_m is the maximum solar heat gain in W/m^2 through glass for wall facing or horizontal roof for July at $40^{\circ}N$ latitude.

The instantaneous heat transfer rate at the inside surface of a wall or ceiling is computed from eq. (5-11):

$$q_i = U\ A\ (TETD) \qquad (5-11)$$

where U is the overall heat transfer coefficient in W/m^2K.

5.2 Infilteration and ventilation

5.2.1. Infilteration

The outdoor air normally has a different heat content than the air within the conditioned space, and therefore impose a thermal load on the air conditioning equipment. Infilteration of air and particularly moisture into a conditioned space is frequently a source of sizable heat gain or loss. The quantity of infilteration air varies according to the tightness of doors and windows, porosity of the building, direction and velocity of wind, and the amount of ventilation and exhaust air.

In low buildings, air infilterates through open doors and cracks on the windward side unless sufficient outdoor air is introduced through the air conditioning equipment to offset it.

The infilteration may be estimated using the crack method or by area of the window or door.

The area method gives the following rates:

	(m^3/s per m^2)
Projected windows with 45% openable area	0.001
Louvre windows	0.011
Wooden doors on one wall in light usage	0.005

The data is based on a wind velocity of 3.3 m/s blowing directly at the window or door, and on observed crack widths and around typical windows and doors. When the direction of wind is oblique, factor of 0.6 is used.

5.2.2. Ventilation

The introduction of outdoor air for ventilation of conditioned space is necessary to dilute the odours given off by people, smoking and other internal contaminants. The amount of ventilation required varies with the total number of people, the ceiling height and the number of people smoking.

The amount of outdoor air has a cosiderable effect on the total refrigeration load and the selection of the dehumidifier.

Some methods of indicating the amount of ventilation are as follows:[9]

a) 25% of total supply air
b) 25% of dehumidified air
c) 0.002 to 0.007 m^3/s per occupant
d) 0.001 m^3/s per square meter of floor area
e) minimum of one air change per hour

5.3 Internal heat gains

5.3.1. People

Human beings give off heat at a metabolic rate which depends on their rate of working. The sensible and latent proportion of the heat liberated for any given activity depends on the ambient dry bulb temperature; the lower the dry bulb temperature the larger the portion of the sensible heat dissipated. For a room dry bulb temperature

of 25°C and for an adult seated and doing light work, the average metabolic rate is 135 W. The sensible portion is 73.5 W and the remainder 61.5 W latent.

5.3.2. Lights

Lights generate sensible heat by the conversion of the electrical power input into light and heat. The portion initially radiated as light eventually becomes heat. For fluorescent lights wattage is multiplied by 1.25 to include heat gain in ballast.

5.4 Duct heat gains

Whenever the air inside the duct is at a temperature different from the ambient temperature, there will be a heat exchange. This may be important if it is of appreciable magnitude because:

i) transmission from a space not being treated but through which the duct passes is a loss of cooling effect

ii) transmission from space being conditioned may put too large a part of cooling effect where it is not wanted

The correction in the first case is either insulation (or dead air space) or greater investment in cooling capacity. In the second, a redistribution of the air to the various supply grilles may be used to compensate for the cooling effect of the duct surface.

CHAPTER 6

HEAT GAIN CALCULATIONS

6.1 Calculation of solar radiation

The amount of solar radiation falling on the walls and roof are calculated from the horizontal surface irridiance, shown in Table 6.1 for 17th March 1978, using the following equations:[7]

$$I_s = I_d \sin\alpha \sin\beta + \sin\beta \cos\alpha \cos n \qquad (6\text{-}1)$$

$$\sin\alpha = \sin d \sin L + \cos d \cos L \cos h \qquad (6\text{-}2)$$

$$h = (T/24)\,360^{\circ} \qquad (6\text{-}3)$$

where

I_d is the direct solar intensity normal to sun's rays
α is the altitude of the sun
β is the angle of the surface with the horizontal
n is the angular difference between the solar azimuth and the wall orientation
L is the latitude of the place
d is the declination of the sun
h is the hour angle
T is the suntime in hours before or after noon

On March 17th the declination for Samaru ($L=11^{\circ}10'$) is $-2^{\circ}03'$ and the surface radiations are calculated and shown in Table 6.2. The surface radiation is required for the modification of TETD which are available for latitude $40^{\circ}N$.

6.2 Modified TETD for walls and roof

For a 12-hour operation of the air conditioning system and applying modification for different latitude and design conditions, the TETD values are obtained. These are shown in Table 6.3.

Walls are assumedto be light coloured having a coefficient of absorbance of 0.5.

Using these TETD values heat gains are calculated by multiplying with area and transmission coefficients.

The building to be air conditioned is divided into 9 rooms for the purpose ofcalculating heat gains and cooling loads. Table 6.4 shows solar heat gain for room (1) calculated at 2 hours interval from 6 to 16 hours.

In a similar manner solar heat gains for the remaining rooms are calculated; the summary of solar heat gains for rooms (1) to (9) is shown in Table 6.5.

6.3 Infiltration and ventilation load

In air conditioning it is more useful to refer to the volume rate of flow Q (m^3/s) rather than the mass flow. The heat content per m^3/s of air is calculated as follows:

sensible hat content

$$q_s = p\, c_p Q\, (t_s - t_r) \quad kJ/s \qquad (6\text{-}4)$$

For standard air conditions of 20°C and atmospheric pressure

$p = 1.20$ kg/m^3

$c_p = 1.01$ kJ/kg °C for dry air

$c_p = 1.89$ kJ/kg °C for water vapour

Allowing for a variation in the supply (or outside in the case of infiltration) temperature (t_s)

$$q_s = 360\ Q\ (t_s - t_r)/(\ 273 + t_s\ ) \qquad (6\text{-}5)$$

The latent heatcontent is given by:

$$q_l = 864{,}000\ Q\ (g_s - g_r)/(\ 273 + t_s\ ) \qquad (6\text{-}6)$$

From indoor design conditions

room dry bulb temperature $t_r = 26^{\circ}C$

room absolute humidity $g_r = 0.0105$ kg/kg of dry air

6.3.1 Infiltration load

The infiltration is estimated for the rooms by the area method as shown in Table 6.6.

The infiltration load at 16 hours is the same as at 14 hours, and assumed to be negligible at 6 hours.

6.3.3 Ventilation load

The ventilation load is calculated using ventilation rate of 0.007 m^3/s per personand assuming occupancies of rooms as shown in Table 6.7. The ventilation rate satisfies the requirement of one airchange per hour for each room.

Each load consists ofthe sensible part followed by its latent component. It should be noted that the ventilation load forms part of refrigeration load.

6.4 Internal heat gains

6.4.1 People and lighting load

The number of peopleoccupying the rooms is assumed the same as for ventilation loadcalculation and a continuous occupancy is assumed. The fluorescent lights are also

assumed to be working continuously, each light emits 80 watts of power. The calculation for each room is tabulated in Table 6.8.

6.5 Duct heat gains

Insulation is necessary whenever the duct surface temperature is lower than the maximum dew point which may occur in the room. Insulation should be used on all ducts installed outside of the building and on cool-air supply ducts extending through unconditioned space.

For a bare sheet metal duct, the surface temperature t_s is given by:

$$t_s = t_o - U(t_o - t_i)/f_o \qquad (6\text{-}7)$$

where

U is the overall transmission coefficient

f_o is the surface conductance of the outside duct surface.

Table 6.1 Horizontal surface radiation obtained at Samaru on 17th March 1978

Hour of day	Surface radiation W/m^2 Total	Direct	Diffuse
6	3	0	3
8	249	130	119
10	740	557	183
12	978	748	194
14	788	601	187
16	441	275	166
18	59	15	44

Table 6.2 Surface radiation on different exposures

Hour of day	Exposure North	East	South	West	Horizontal
6	3	3	3	3	3
8	28	421	33	28	249
10	50	379	86	50	740
12	55	55	111	55	978
14	53	53	87	403	788
16	47	47	55	700	441
18	13	13	13	255	59
R_s	55	421	111	700	978
R_m	77	529	223	529	752

Table 6.3 TETD in °C for walls and roof[4]

Hour of day	TETD °C 6	8	10	12	14	16
Cement block wall 400 mm						
Exposure North	9.4	8.3	6.7	5.0	6.1	7.2
East	10.8	12.1	18.4	16.4	9.6	8.3
South	10.9	8.3	6.8	7.1	8.8	9.6
West	28.4	27.3	12.4	8.9	10.7	18.1
Cement block wall 200 mm						
Exposure North or shade	9.4	6.7	4.4	5.0	7.8	9.4
Roof 200 kg/m^2	26.1	21.5	15.4	18.1	22.4	25.1

Table 6.4 Solarheat gain for room (1)

	Area m^2	U W/m^{2}K	Hour of day 6	8	10	12	14	16
Exposure	Wall		(q in W)					
North	23.50	0.32	71	62	50	38	46	54
East	39.93	0.43	161	115	76	86	134	161
South	23.50	0.32	82	62	51	53	66	72
West	42.70	0.32	388	373	169	127	146	247
Glass	40.52	2.02		180	360	589	769	769
Door	2.77	0.93		3	10	18	24	24
Roof	152.50	0.41	1632	1344	963	1132	1401	1569
	Glass	GF						
North	20.26	0.16	10	91	162	178	172	152
South	20.26	0.16	10	107	279	360	282	178
			2354	2337	2129	2561	3040	3226

Table 6.5 Summary of solar heat gains

Room	hour of day 6	8	10	12	14	16
1	2354	2337	2129	2561	3040	3226
2	6040	5817	5584	6636	7634	7858
3	357	499	638	815	890	855
4	346	483	520	618	760	811
5	259	270	317	410	513	521
6	115	167	249	333	393	377
7	115	167	249	333	393	377
8	136	139	169	219	260	258
9	325	673	1164	1448	1516	1393
[redacted]	10047	10552	11019	13373	15126	15676

[redacted] Infiltration load

Room	Infilt. m^3/s	Infilt. load (W) : Hour of day 8		10		12		14	
1	0.028	67	426	146	483	237	419	307	416
2	0.078	188	1186	407	1177	660	1166	856	1158
3	0.008	19	121	42	121	68	120	88	119
4	0.028	67	426	146	423	237	429	307	416
5	0.041	99	623	214	619	347	613	450	609
6	0.041	99	623	214	619	347	613	450	609
7	0.041	99	623	214	619	347	613	450	609
8	0.014	34	213	73	211	119	209	154	208
9	0.037	89	563	193	558	313	553	406	549

Table 6.7 Ventilation load

Room	No.	vent. pers.	m^3/s	Vent. load (W): hour of day 8		10		12		14	
1	20	.149		392	2265	823	2249	1261	2228	1635	2212
2	60	.473		1244	7191	2611	7139	4004	7074	5190	7023
3	10	.073		192	1110	403	1102	618	1092	801	1084
4	10	.073		192	1110	403	1102	618	1092	801	1084
5	5	.037		97	563	204	558	313	553	406	549
6	5	.037		97	563	204	558	313	553	406	549
7	5	.037		97	563	204	558	313	553	406	549
8	2	.015		39	228	83	226	127	224	165	223
9	20	.149		392	2265	823	2249	1261	2228	1635	2212

Table 6.8 People and lighting load

Room	No. of people	People load (W) Sensible	Latent	No. of lights	lighting load sensible (W)
1	20	1470	1230	12	1200
2	60	4410	3690	36	3600
3	10	735	615	9	900
4	10	735	615	9	900
5	5	368	308	4	400
6	5	368	308	4	400
7	5	368	308	4	400
8	2	147	123	4	400
9	20	1470	1230	12	1200

CHAPTER 7

COOLING LOAD

The cooling load is the rate at which heat energy must be removed from the conditioned space to maintain the temperature and humidity at the design value. The cooling load will generally differ from the heat gain since some of the heat gains, such as solar radiation, are absorbed by the structure and the contents and do not appear as cooling load until sometime later. Interior surfaces and other objects are cooled by convection when they attain a temperature higher than room air. The heat storage characteristics of the structure governs the relationship between heat gain and cooling load. In general radiant heat transfer inputs to the structure are delayed while convective and latent inputs are not. The reduction in the peak load due to storage effect can be utilised in selecting an economical cooling equipment.

7.1 Thermal storage

A major portion of the instantaneous heat gain is in the form of radiant heat absorbed by the solid surfaces of the space. The portion transferred into the air becomes part of the instantaneous cooling load while the remainder is stored and released later.

If the cooling system operates for only part of the day (say 12 hours), some of the heat remains within the

structure and is removed when the equipment is turned on the next day, appearing as a pull-down load. The magnitude of the pull-down load increases as the hours of shutdown increases.

Thus the peak cooling load is substantially less than the peak instantaneous gain and lags by a length of time which increases with the mass of the heat storing material.

The ASHRAE Handbook outlines two methods of computing the cooling load from the heat gain. One method depends on detailed hourly computations of the heat gains using the transfer function method. This method is currently the most reliable and has the advantage of furnishing information necessary to compute energy requirements. However, because of the scope of the necessary computer program, the method is generally not used except for large commercial design problems.

A simplified method for relating the heat gain to the cooling load is used for residential, light commercial and small scale design projects. This method utilizes an averaging technique and requires experience and judgement on the part of the designer. The general procedure is:
1) Seperate the instantaneous heat gains into convective and radiant components using Table 7.1. In this context radiant refers to the portion of heat gain that is absorbed and released later. The convective portions are assumed to make up an instantaneous cooling load while the radiant components are absorbed and later appear as cooling load.

Latent heat gains are assumed to be instantaneous.

2) Average the radiant components over a period of time immediately preceding the selected time. The time period depends on the weight of the structure. This varies from 2 hours for light construction to 8 hours for heavy construction.

3) Sum the convective and averaged radiant components to obtain the instantaneous cooling load.

7.2 Room cooling loads

Cooling load is calculated at 14 hours when solar load is maximum and using a 4 hour averaging period. This is shown in Table 7.2 for room (1).

The room sensible heat factor (RSHF) is the ratio of the room sensible heat to the total heat load and is calculated as:

$$RSHF = \frac{\text{Sensible heat}}{\text{Sensible heat + Latent heat}} \qquad (7\text{-}1)$$

In a similar manner, cooling loads are calculated for the remaining number of rooms and the results are summarized in Table 7.3.

7.3 Room air quantities

The air quantity required to offset simultaneously the room sensible and latent heat loads may be calculated using the RSHF linedrawn on a psychrometric chart and apparatus dew point (ADP). ADP is the effective surface temperature of the apparatus, and can be considered to be a uniform surface temperature which would produce the same

leaving air conditions as the non-uniform surface temperature that actually occurs when the apparatus is in operation.

The room design conditions are

temperature 26°C relative humidity 45 per cent

specific enthalpy 51.5 kJ/kg specific volume 0.87 m^3/kg

This condition is shown asthe point R on the psychrometric chart (Fig.7.1). The outdoor condition is the point O. Assuming a maximum outdoor air quantity of 30 % the mixing condition M is suchthat itdivides the line RO in the ratio 3:7.

Thus M has the conditions

temperature 30°C relative humidity 45 %

specific enthalpy 60 kJ/kg.

A number of points P are selected on the saturation curve on the psychrometricchart and the supply point S is estimated on the line PM assuming a by-pass factor of 0.20. These supply pointsare joined by a smooth curve to give all the possible supply points. One such point is selected to obtain the required room sensible heat factor (0.70), and has conditions

temperature 11.5°C specific enthalpy 32 kJ/kg

specific volume 0.83 m^3/kg.

The cooling produced per m^3/s of air is given by

$$\frac{51.5 - 32}{0.86} = 22 \text{ kW / } (m^3/s)$$

The air quantities for each room are shown in Table 7.4

Table 7.1 Rádiant component of heat gains[10]

Heat gain component	Radiant	Convective	Latent
	Percentages		
Transperant sufaces:			
Without inside shading	100	0	0
with inside shading	58	42	0
Conduction & Convection	0	100	0
Fluorescent lights	50	50	0
People	40	20	40
Infilteration and ventilation	0	sum equals	100
Heat through walls and roof	[illegible]	[illegible]	0

Table 7.2 Cooling load for room (1)

Heat through:walls and roof (W)				windows
hour	Total	convective	radiant(60%)	radiant(100%)
10	1688		1013	441
12	2023		1214	538
14	2586	1034	1552	454
			1260 av.	478 av.

Source	Sensible	Latent
Solar	2772	
People	1470	1230
Lights	1200	
Infilteration	307	416
Total	5749 W	1646 W
RSHF = 5749/7395 = 0.78		

Table 7.3 Summary of cooling loads for rooms (1) to (9)

Room	Cooling load (W) Sensible	Latent	Total	RSHF
1	5749	1646	7395	0.78
2	15918	4848	20766	0.77
3	2557	734	3291	0.78
4	2625	1031	3656	0.72
5	1673	917	2590	0.65
6	1572	917	2489	0.63
7	1572	917	2489	0.63
8	936	331	1267	0.74
9	4521	1779	6300	0.72

Table 7.4 Room air quantities

Room	Cooling load W	air quantity m^3/s
1	7395	0.3430
2	20766	0.9448
3	3291	0.1498
4	3656	0.1663
5	2590	0.1179
6	2489	0.1133
7	2489	0.1133
8	1267	0.0576
9	6300	0.2867

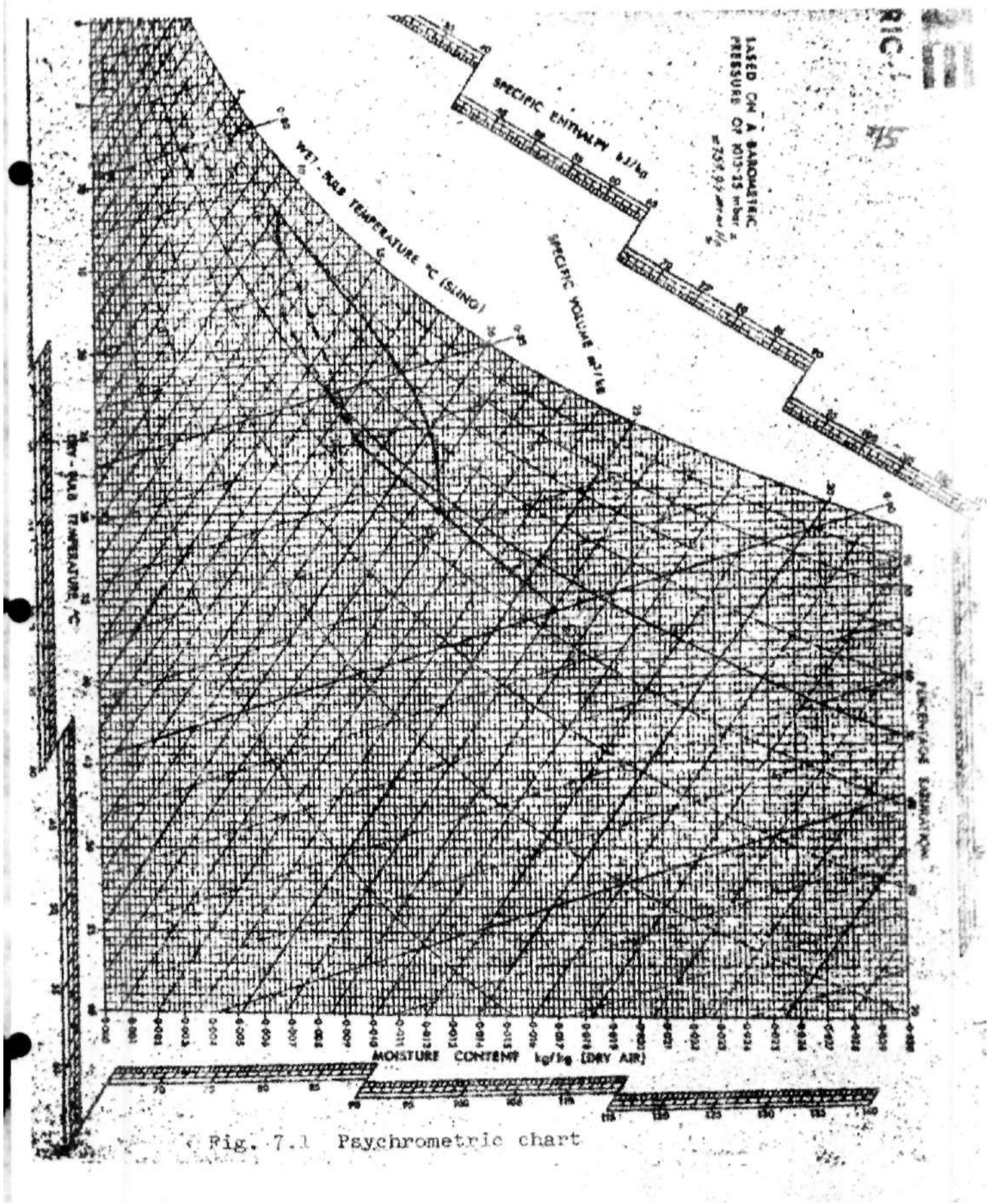


Fig. 7.1 Psychrometric chart

CHAPTER 8

REFRIGERATION AND AIR HANDLING EQUIPMENT

8.1 Refrigeration

Many refrigeration cycles are suitable for air conditioning purposes. These are mechanical vapour compression, vapour thermo-compression and absorption refrigeration cycles. Mechanical compression systems using reciprocating or centrifugal compressors have found the greatest acceptance in air conditioning work. Reciprocating compressors are designed for small and medium size refrigeration system upto 200 kW to be used with refrigerants having low specific volume and relatively high pressure characteristics. Compressors can be either open or hermetic type

A refrigeration machine may be classified as one of the following:

a) A compressor unit consisting of a compressor motor and safety controls mounted as a unit.

b) A condensing unit consisting of a compressor unit plus an interconnected water-cooled or air-cooled condenser mounted as a unit.

c) A water chilling unit consisting of either a compressor unit or a condensing unit, plus an interconnected water cooler and operating controls mounted as a unit.

8.1.1. Refrigeration compressors

There are four main compressor types: reciprocating, centrifugal, screw type and rotary. Only the first two are commonly used in air conditioning work.

Reciprocating compressors are normally single acting and may employ upto sixteen cylinders frequently arranged in V or VW formation. On larger machines capacity reduction is achieved by cylinder unloading. Capacities usually range up to about 425 KW using refrigerant - 12 (R - 12) or 565 KW with R - 22 at a 50 Hz speed of approximately 24 rev per second. Direct drive arrangements are preferable with electrical input above 38 KW. Electric motors may be of the open or hermetic type.

Centrifugal compressors may be single or multistage and are often driven through speed increasing gears with electric motor drive. Open machines are available in capacities upto about 16 000 KW; standard hermetic units range upto around 7 000 KW. Capacity control down to 10 percent or less is achieved by variable guide vanes at the compressor suction inlet.

Compressors may run at between 50 and 300 rev per second. Drive is normally by electric motor though gas and steam turbines, gas and diesel engines are sometimes used. For air conditioning applications refrigerants employed include R-11, R-12, R-113 and R-114.

The screw compressor is a high speed positive displacement type of machine, suitable for most refrigerants, obtaining compression by the interaction of two screw cut rotors. The oil-injected version, available as a hermetic or open machine, is currently manufactured in capacities upto about 1750 KW, and capable of capacity control down to about 10 percent with unloaded starting.

8.1.2. Condensers

The heat rejected by the refrigeration system is removed at the condenser. The cooling medium may be air, or water, or both. Water-cooled condensers may use the water only once, as with river, lake or sea water systems, and mains water to wastemay be permitted with some smaller plants. Generally the water is recirculated, recooled in a cooling tower (or other evaporative water cooling device) and reused. Vapour-compression systems use about 0.053 kg per second per kW at 5.6°C water temperature rise.

Air cooled condensers are usually coils of finned copper tube construction with aluminium or copper fins, over which ambient air is circulated by a fan. All-aluminium construction is also now available. The condensers are generally constructed as single units, and built up into banks for higher capacities.

8.2 Air conditioning apparatus

Air conditioning or handling apparatus uses either an air washer or cooling coils for the purpose of achieving the desired change in the air properties. The principal difference between the two types of equipment, beside their construction, is that the air washer gives a greater flexibility in operation. It can be used to humidify the air very easily, although sprayed cooling coils are available for humidification purposes.

Due to water in the sump tank of the air washer (or sprayed cooling coils) the thermal inertia is greater than for the cooling coils units, and this will result in stable conditions for the air leaving the washer. Such advantages have to be weighed against the disadvantages; the washer is bulky, the corrosion risk is greater than for cooling coils, maintenance is generally more expensive and more importantly the efficiency is generally lower due to less effective heat transfer.

8.2.1. Air washer

Air washer consists of a rectangular steel chamber closed at the top and sides and mounted on a shallow watertight tank of steel or concrete. Inlet baffles located at the air entering end of the washer promote uniform air velocities through the washer and minimize the spraying back of water into the entrance chamber as a result of air eddy currents. At the air leaving end of

the washer eliminators are provided to remove entrained water droplets.

Within the washer spray chamber two banks of opposing spray nozzles provide fairly divided droplets of water uniformly distributed. After contacting the air, the water is collected in the tank and is returned to the sprays by a recirculating pump.

The relationship between the initial air state and the spray water temperature determines whether heating, cooling, humidification or dehumidification of air results. If the spray water is recirculated until equilibrium conditions exist, adiabatic saturation occurs and the water assumes the wet bulb temperature of the entering air, which has its humidity increased as its dry bulb temperature is reduced. Heating the spray water will promote humidification accompanied by an increase or decrease in the leaving air dry bulb temperature.If the water is cooled before spraying, the air will be cooled, but whether humidification or dehumidification takes place depends on whether the entering air iscooled below its dew point temperature or not.

Washer efficiency increases slightly with higherspray spray pressure, reduced air velocity, increased water quantity per unit face area, increased washer length and increase in washer height above the tank.

Most washer ratings are based on an average velocity of 2.5 m/s, as velocities above 3.8 m/s and below 1.8 m/s often result in faulty elimination of entrained moisture. The quantity of water sprayed is generally about 0.075 to 0.110 kg/s per nozzle, the nozzle spacing being arranged to give 0.24 to 0.63 kg/s per m^3/s of air. The fineness of the spray depends on the nozzle design and spray pressure, which is commonly between 1.7 and 2.6 bar.

Washers vary in length from 1.5 to 3.0 m according to the design and number of spray banks. A spray may assist or resist the flow ofair depending on its direction, the kinetic energy in the atomised spray amounts to approximately 25 N/m^2 for a water loading of 0.61 kg/s at 2.5 m/s face velocity. Overall washer resistance may therefore vary from about 50 N/m^2 to 125 N/m^2.

8.2.2. Cooling coils

Coils for the heating or cooling of air are constructed of plain tubes or piping having a finned surface bounded to, or forming a part of the coil tubing. Extended surfaces are employed to increase the external surface heat transfer area.

The transfer of heat from air flowing over a pipe to a fluid flowing within it is impeded by three resistances. First, from the air to the surface of the pipe, known as the outside film air resistance. Seconly, the resistance to conduction of heat through the pipe wall itself;

and thirdly, the inside surface (or fluid film) resistance between the inner surface of the pipe wall and the fluid flowing in the pipe.

In air conditioning, heating and cooling applications, the resistance of the pipe wall and the inside film resistance are low compared with the outside film resistance. It is therefore advantageous, in terms of economies in space, weight and cost, to decrease the outside film resistance; and this is most readily achieved by increasing the external surface by means of fins.

The cooling and dehumification coil is designed to match the sensible and latent heat requirements of a particular job and that the fan is sized to handle the required volume of air. Specially constructed equipment cannot be justified for small commercial and residential applications. Furthermore these applications generally have a high sensible heat factor and dehumidification is not as critical as it is in large commercial buildings. Therefore the equipment is manufactured to operate at or near one particular set ofconditions. For example, typical residential and light commercial cooling equipment operates operates with a coil SHF of 0.75 to 0.80 with coils having tubes of 9.5 to 19.0 mm outer diameter and wall thickness of 0.38 to 0.89 mm. Fins are 0.15 to 0.43 mm thick and spaced at the rate of 3 to 14 fins per 25 mm.

Center line spacing of tubes and rows varies from about 25 to 45 mm. Resultant surface ratios (outside to inside surface area) range from 5 to 50 with ratios of 6 to 60 for outside surface to air surface area.

Air side performance is affected primarily by air contact and air friction. By-pass factor is important in determining the extent of reduction of the state of the air towards the apparatus dew point. In addition to the basic variables of the face velocity, arrangement of tubes and fins, and the coil depth in rows, the method of baffling around the sides and ends of coil is of considerable importance in preventing air leakages around the cooling surface.

8.3 Central air conditioning system

The Westinghouse ST020 selected for this design is an air cooled condensing unit consisting of two semi hermetic compressors enclosed in weatherproof galvanized steel panels of gauge 20. Condenser cooling fans are of propeller type and three in number, 400 mm in diameter each and driven by 560 W motors through adjustable V-belt drives.

The air handling unit (Westinghouse AB060) has a nominal flow rate of 2.8 m^3/s with a nominal cooling capacity of 69,380 W. The flow rate can be adjusted in the range of 2.33 to 3.83 m^3/s. The cooling coil has two equal circuits each consisting of four rows of tubes having outside diameter of 12.7 mm. Fans are forwardly curved blades, centrifugal type and are driven by adjustable V-belt drives.

CHAPTER 9

AIR DUCT DESIGN

The function of a duct system is to transmit air from/to the air handling apparatus to the space being conditioned. To fulfil this function in a practical manner, the system must be designed within the prescribed limits of available space, friction loss, velocity, sound level, heat and leakages losses and gains. In an air conditioning system the cost of ducts may be as much as 25 per cent of the total investment. The resistance of the duct system is a substantial portion of the static pressure against which the fan operates.

Supply and return air duct systems are classified according to the velocity and pressure within the duct. With respect to the initial velocity the duct systems can be divided into:

Low velocity system : velocities upto 12.7 m/s
normally between 6.1 and 11.2 m/s.

High velocity system : velocities above 12.7 m/s.

Normally return air systems for both low and high velocity supply air systems are designed as low velocity system.

Air distribution systems can be further divided into three pressure categories as follows:

1) Low pressure : upto 930 N/m^2 : Class I fan

2) Medium pressure : 930 to 1680 N/m^2: Class II fan
3) High pressure :1680 to 3048 N/m^2: Class III fan

These pressure ranges are total pressures, including the losses through the air handling apparatus, ductwork and the air terminals in the space.

9.1 System resistance

The system resistance consists of
i) resistance of inlet and filters
ii) air handler pressure drop
iii) resistance of supply ducts and terminals, and,
iv) resistance of return ducts and grilles.

Direct drive fans are used on applications where system resistance can be accurately determined. However, most air conditioning applications use belt drives.

V-belts must be applied in matched sets and used on balanced sheaves to minimize vibration problems. They are particularly useful on applications where adjustment may be required to obtain more exact air quantities. These adjustments can be accomplished by varying the pitch diameter on adjustable sheaves, or by changing one or both sheaves on a fixed sheave drive.

9.2 Duct design methods

In the design of air duct systems four methods are employed, namely velocity reduction, equal friction, static regain and digital computer methods. The first three methods and their refinements represent different design levels of accuracy and complexity, and they should be selected to suit the application.

9.2.1. Velocity reduction method

This method consists of selecting the velocity at the fan discharge and designing for progressively lower velocities in the main duct at each branch duct. With the selected velocities and known air flow rates, the various duct diameters are read directly from tables, and the equivalent rectangular sizes are obtained. The pressure loss of the run having apparently the highest resistance is determined by adding the straight pipe, elbow and transmission losses. This total value represents the fan static pressure required for the supply duct system. The return air system is sized similarly, starting with the lowest velocities at the return intakes and incresing them progressively in the direction of the fan inlet. Dampers are relied upon for balancing the system.

A refinement of this method is to size several branch ducts to dissipate the pressure available at the entrance to each. The pressure loss of the ductwork between the fan and first branch takeoff is subtracted from the now

known fan static pressure to obtain the available pressure at each junction. By trial, a branch velocity is found that results in the branch pressure loss being equal to, or somewhat less than, that available. The procedure is repeated for each branch.

If the fan is specified so that the static pressure available for the ductwork is known, the method consists of finding by trial, the velocities in the main duct that will result in a pressure loss equal to the pressure available.

The merits of the velocity reduction method are

1) duct sizes are determined very easily, and
2) velocities can be limited to those known to be safe from causing noise problem

Its weaknesses are

1) the designer cannot always determine by inspection which run probably has the highest resistance
2) proper choice of velocities requires experience and judgement.

9.2.2. Equal friction method

The principle of this method is to make the pressure loss per unit length the same for the entire system. With this method little balancing is required for symmetrical layouts in which all runs have the same resistance. For layouts having both short and long runs, the shortest run will require considerable dampering.

A modification of this method would be to design the longest run at two or more different friction rates. If due to space limitations, a high friction rate must be used on the upstream side of the duct system, this rate can be reduced when space conditions become less critical on the downstream part of the system.

Usual practice is to select the velocity in the main duct near the fan from the standpoint of noise for particular application. Since the flowrate is known this establishes a value of friction loss per unit length. This friction loss is maintained throughout the design. An advantage of this method is that it automatically reduces the duct velocities in the direction of the flow, thereby ensuring that the problem of noise will be considerably reduced.

After sizing the system the pressure loss of the run having apparently the highest resistance is calculated. In so doing, the pressure losses of all elbows and transitions are included, and expressed in terms of equivalent length of straight duct. The principal limitation of equal friction method is that it does not differentiate between runs having several transitions, elbows etc. and runs having none. Only the actual length of the duct is considered; this and the flowrate fix the duct size. Moreover, when computing the system resistance, care must be exceci-sed that the pressure losses of transitions, elbows etc., are included and added to the straight duct losses.

If the pressure available for the ductwork is known the pressure can be divided by the total equivalent length of the run apparently having the highest resistance to obtain a design friction loss value per unit length. Hence it is not necessary to select an initial velocity. However the method then has the weakness that the resistances of fittings must be expressed in terms of equivalent lengths. Since transitions, elbows etc., have predominantly dynamic losses, the equivalent length of a particular fitting varies considerably with its actual size. Hence, when the available pressure is known the method requires that the duct size be estimated in advance. The calculated duct size should therefore be compared with initial estimate and if considerably different, a recalculation should be made using the calculated size.

9.2.3. Static regain method

The principle of the static regain method is to size a duct run so that the increase in static pressure (regain) at each takeoff junction just offsets the pressure loss of the succeeding section of the run.

The method provides a convenient means of designing a long run of duct having several takeoffs so that essentially the same static pressure exists at the entrance to each branch, outlet or terminal takeoff. As a consequence, outlet or terminal selection and system balancing is simplified. The method is particularly suited to large

installations having several long runs of duct, with each run having many takeoffs or supply outlets attached. For this type of application little or no dampering is ordinarily required to balance the system.

The duct length of any section should include the equivalent length of any elbow or transition within the section. If no friction or dynamic losses occured at the junction, there would be no loss in the total pressure, and the change in velocity pressure would be completely converted into a regain (rise) in static pressure. It has been found by test that the regain coefficient across a takeoff or an outlet will equal 0.90 for a well designed and constructed duct, with no reducing section immediately after the outlet. However for practical design an average recovery of 0.75 is assumed.

9.2.4. Digital computer method

The applications of computers to ductwork design originated in mining engineering. Analogies exist between the various classified forms of energy flow. Kirchhoff's laws applicable to any energy distribution network state that neglecting leakage and the action of external agents

a) the same quantity of air enters a junction as leaves it

hence $$\sum_{i=1}^{n} Q_i = 0 \qquad (9\text{-}1)$$

b) the total pressure drop in a closed circuit is the algebraic sum of the pressure drops of the individual pathways, and is equal to the algebraic sum of the

pressure differentials $\triangle p$, established by the fans,

hence $$\sum_{i=1}^{n} c_i Q_i^2 = \triangle p \qquad (9\text{-}2)$$

but if no fans are present then $\triangle p = 0$.

Using the expansion of the basic flow law, thus

$$\triangle p = c\,(Q + x)^n \qquad (9\text{-}3)$$

where Q is an approximation of the volume flow rate,

x is the error in the flow rate, and

n is an index of flow

and since $x \ll Q$, the terms containing higher powers of x may be neglected, hence

$$\triangle p - cQ^n = (nc/1!)\, Q^{n-1} x \qquad (9\text{-}4)$$

or the correction

$$x = (\triangle p - cQ^n)/(ncQ^{n-1}) \qquad (9\text{-}5)$$

A series of successive approximations result in the correct value of Q. This technique is applied to all the sub-circuits of the network.

9.3 Losses in ducts

9.3.1. Friction loss in ducts

Along an ideal frictionless duct, the total pressure being the sum of static and velocity pressures remains constant. In actual duct system pressure losses occur due to two effects: friction losses primarily from surface friction; and dynamic losses from relatively sudden changes in velocity or direction or other eddy sources.

Most of the pressure drop in a straighr duct is caused by surface friction. The fanning equation (9-6) relates frictional pressure drop in a duct to an emperical dimensionless friction factor.

$$h_f = f \left(\frac{L}{R_h} \right) \rho \frac{V^2}{2g} \qquad (9\text{-}6)$$

where f is the fanning friction factor

R_h is the hydraulic radius

The pressure loss due to friction is greater for a rectangular duct than a circular duct of the same cross-sectional area and capacity. For most practical purposes ducts of aspect ratio not exceeding 8,will have the same loss for equal length and mean velocity of flow as a circular duct of the same hydraulic diameter. When the duct sizes are expressed in terms of hydraulic diameter and when the equations for friction loss in round and rectangular ducts are equated for equal lengths and capacity Eq. (9-7) for the circular equivalent of a rectangular duct is obtained.

$$D_e = 1.3 \frac{(ab)^{\frac{5}{8}}}{(a + b)^{\frac{1}{4}}} \qquad (9\text{-}7)$$

where a and b are the sides of rectangular duct.

The pressure lost due to friction in a straight constant area duct is given in Fig. 9.1 for air flowing in galvanized steel ducts with approximately 40 joints per 30 m. This chart is based on standard air and fully

developed flow, and can be used for a temperature range of 10°C to 38°C without any corrections for viscosity and density changes.

9.3.2. Dynamic loss in fittings

Whenever a change in area or direction occurs in a duct or when the flow is divided or diverted into a branch, a substantial lossin total pressure may occur. These lossessare generally of greater magnitude than losses in the straight duct. Dynamic losses vary substantially as the square of the velocity and are conveniently represented by Eq. (9-8).

$$h_o = C\, h_v \qquad (9\text{-}8)$$

where the loss coefficient C ia a constant

Fittings are classified as either constant flow such as an elbow or transition or as divided flow such as a wye or tee. Elbows are generally efficient fittings in that their losses are small when the turn is gradual. When an abrupt turn is used without turning vanes the pressure loss will be four to five times larger. Table 9.1 gives loss coefficients C for a tee joint, at various ratios of V_1 to V_2 (V_2 being perpendicular to V_1).

Dynamic losses for round and rectangular radius elbow are given in terms of additional equivalent lengths (L/D) as shown in Table 9.2

Duct transformations are used to change the shape of a duct or to increase or decrease the duct area. When the shape of a rectangular duct is changed but the cross-sectional area remains the same a slope of 1 in 7 is recommended for the sides of the transformation piece; if this slope cannot be maintained a maximum slope of 1 in 4 should not be exceeded. Often ducts must be reduced in size to avoid obstructions. It is good practice not to reduce the duct more than 20 percent of the original area. Accepted methods of duct design usually indicate that a reduction of at least 50 mm can be made; however, it is recommended that the original duct size be maintained. Savings in installed cost of as much as 25 percent can be realized by running the duct at the same size for several terminals.

All duct sizes should be in multiples of 50 mm and reductions should be in one dimension only. The recommended minimum duct size is 200 mm x 250 mm for fabricated shop ductwork.

Table 9.3 shows loss coefficients for three transformations pieces, namely gradual expansion and contraction, and equal area transformer.

9.4 Air distribution devices

Air distribution devices include various types of grilles, diffusers and special devices such as integrated air/light units and perforated ceilings.

Air emerging from an outlet entrains room air. This induction effect increases the volume and decreases the velocity of the mixture stream according to the conservation of momentum. Room air near the jet is entrained and is replaced by surrounding air and thus all the room air is set into motion. Any obstruction that limits the access of secondary air to the primary stream will decrease the induction and lengthen the throw. This effect occurs when the outlet stream is released parallel and close to a smooth ceiling or wall. The absence of induction on one edge increases the throw by 25 to 50 percent. Unless greater part of the heat load is at the opposite wall, it is advisable to select a throw equal to 75 percent of the distance to the opposite wall. Whenever the average room air velocity is less than 0.25 m/s buoyancy effects may be significant. In general a velocity of 0.08 m/s is required to prevent stagnant regions. The region within 0.3 m of the wall and above 1.8 m from the floor is not normally considered to be in the occupied zone.

Hence outlets are selected to provide adequate room air distribution in terms of throw, the angle of spread, and noise criteria.

Noise Criteria (NC) numbers are used for providing information on the spectrum content of noise. These are a series of curves constructed using loudness contours, and speech-interfering properties of noise and are used as a simple means of specifying sound level limits for an environment by a simple, single number rating. They have been found to be quite generally applicable for conditions of comfort. In general, levels below a NC of 30 are considered to be quiet, NC of 30 to 40 is recommended for libraries and classrooms and levels above NC 50 are considered noisy.

9.4.1. Sidewall Outlets

1) Perforated grille

This grille has a small vane ratio (usually from 0.05 to 0.20) and, therefore, has little directional effect. Consequently, it is used principally as an exhaust or return grille but seldom as a supply grille. When mannual shut-off damper backs up this grille it becomes a register.

2) Fixed bar grille

The fixed bar grille is used satisfactorily in locations where flow direction is not critical or can be predetermined. A vane ratio of one or more is desirable. To obstruct the line of sight into the duct interior, closely spaced vanes are preferred.

3) Adjustable bar grille

This grille is the most desirable for side wall location. Since it is available with both horizontal and

vertical adjustable bars, minor air motion problems can be quickly corrected by adjusting the vanes.

4) Slotted outlet

This outlet may have multiple slots widely spaced, resulting in about 10 percent free area. Performance is about the same as for a bar grille of the same flow rate and static pressure, but the blow is shorter because of greater induction at the outlet face. Another design to effect early completion of induction is the long single, or double, horizontal slot. It is particularly advantageous where low ceiling heights exist and outlet height is limited, or where objections to the appearance of grilles are raised.

Fig. 9.3. shows the basic room air motion and temperature profile for a single side wall outlet.

9.4.2. Ceiling outlets

1) Pan outlet

This simple design of ceiling distribution makes use of a duct collar with a pan under it. Air passes from the plenum through the duct collar and splashes against the pan. The pan should be of sufficient diameter to hide the duct opening from sight, and also should be adjustable in distance from the ceiling. Pans may be perforated to permit part of the air to diffuse downward. Advantages of the pan outlet are low cost and ability to hide the air opening. Disadvantages are lack of uniform air direction because of

poor approach conditions and the tendency to streak ceilings.

2) Ceiling diffuser

These outlets are improvements over the pan type. They hasten induction somewhat by supplying air in multiple layers. Approach conditions must be good to secure even distribution. Frequently they are combined with lighting fixtures, and are available with an internal induction feature.

3) Perforated ceilings and panels

Various types of perforated ceilings for the introduction of conditioned air for comfort and industrial systems are available. The principal feature of this method of handling air is that a greater volume of air per square metre of floor area can be introduced at a lower temperature with a minimum of movement in the occupied zone and with less danger of draft. Since discharge velocity is low, induction is low. Therefore,care must be taken to provide adequate room air motion in excess of 0.075 m/s.

Fig. 9.4 shows the basic room air motion and temperature profile for a single ceiling diffuser.

9.5 Duct construction

Sheet metal continues to be in general use for duct-work fabrication, but the possibility of using other materials must not be forgotten. Assuming that the materials are readily available, the various factors which govern the choice of materials are:

a) weight - a light weight duct structure aids handling and makes transportation easier.

b) flexibility - flexible tubing assists installation in tight spaces; joining is also easier.

c) fire risk - the material should fulfil the fire regulation and an adequate number of fire dampers must be installed in the distribution system.

d) corrosion - the ductwork must be able to withstand water corrosion.

e) thermal conductivity - any heat transfer through the duct walls should be minimum; insulation may be required when the air stream and ambient temperature differential is large.

f) friction factor - the surface roughness of the duct wall should be low.

g) noise - metal readily conduts sound; the vibrational energy is radiated from the duct walls as sound to the surroundings.

h) static electricity -due to dielectric properties materials (such as plastics) allow the free electron charge density in thestream to build up.

i) cost - a balance must be achieved between the capital, maintenance and amortization costs and the required properties ofthe ductwork.

Plastics, fibre-glass, various kinds of flexible ducts, asbestos cement and builders' work ducts can offer some advantages in the above properties as compared with sheet metal.

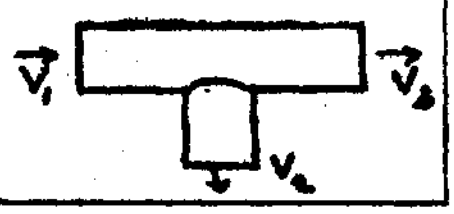
The installed or first cost of the ductwork depends on the amount of material used and the difficulty experienced in fabricating the ducts. The duct construction class,see Table 9.4, can be used asa measure of cost.

Table 9.5 shows the recommended sheet-metal gauges for galvanized steel ducts.

Ducts and sheet metal connections may be fabricated according to thefollowingmethods of construction as shown in Fig.9.5:

a) drive slip
b) S-slip
c) inside groove seam
d) bar slip
e) sliding seam
f) pocket joint section at clip punch
g) standing seam
h) Pittsburghseam

Table 9.1 Loss coefficients C for a tee joint.

V_2/V_1	C	Tee joint
0.2	4.00	
0.5	2.00	
1.0	1.75	
5.0	1.50	

(Loss referred to V_2)

Table 9.2 Dynamic losses in elbows.

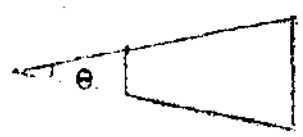
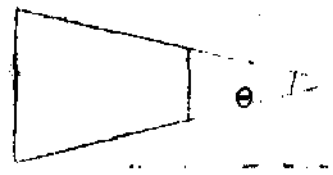
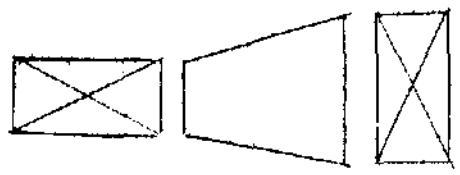
Round radius elbow*	R/D 0.5	0.75	1.0	1.5	2.0
	L/D ratio 45	23	17	.12	10
Rect-angular elbow W/D = 1	R/D 0.5	075	1.00	1.25	1.50
Vaned					
No.of Vanes					
1	18	10	8		7
2	12	8	7		7
3	10	7	7		6
Unvaned W/D					
0.5	33	14	9	5	4
1	45	18	11	7	4
3	80	30	14	8	5
6	125	40	18	12	7

* See Figure 9.2

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Table 9.3 Loss coefficient for area changes

Transformation	Angle θ°	$^{*}C_r$	C_2
 Expansion	5	0.17	
	10	0.28	
	20	0.45	
	30	0.59	
	40	0.73	
 Contraction	30		0.02
	45		0.04
	60		0.07
 Transformer	$A_1 = A_2$		0.15

* Loss coefficient for gradual expansion is $C_r \times C_{1a}$ where C_{1a} is loss coefficient for an abrupt expansion

$$C_{1a} = 1 - (V_2/V_1)^2 \qquad (9\text{-}9)$$

Table 9.4 Duct classes

Duct class	maximum side (m)	semi perimeter (m)
1	0.150 - 0.445	0.255 - 0.585
2	0.305 - 0.610	0.610 - 1.170
3	0.610 - 1.015	0.815 - 1.170
4	0.660 - 2.235	1.220 - 2.390
5	1.220 - 2.285	2.440 - 4.470
6	2.285 - 3.660	2.440 - 6.045

Table 9.5 Recommended Sheet-Metal Gauges

Gauge (U.S.Std)	Thickness (mm)	Weight, kg/m^2	Rectangular duct max side (m)	Round duct diameter (m)
26	0.56	4.42	upto -0.305	upto-0.483
24	0.71	5.64	0.306-0.762	0.483-0.737
22	0.84	6.86	0.763-1.524	0.738-0.991

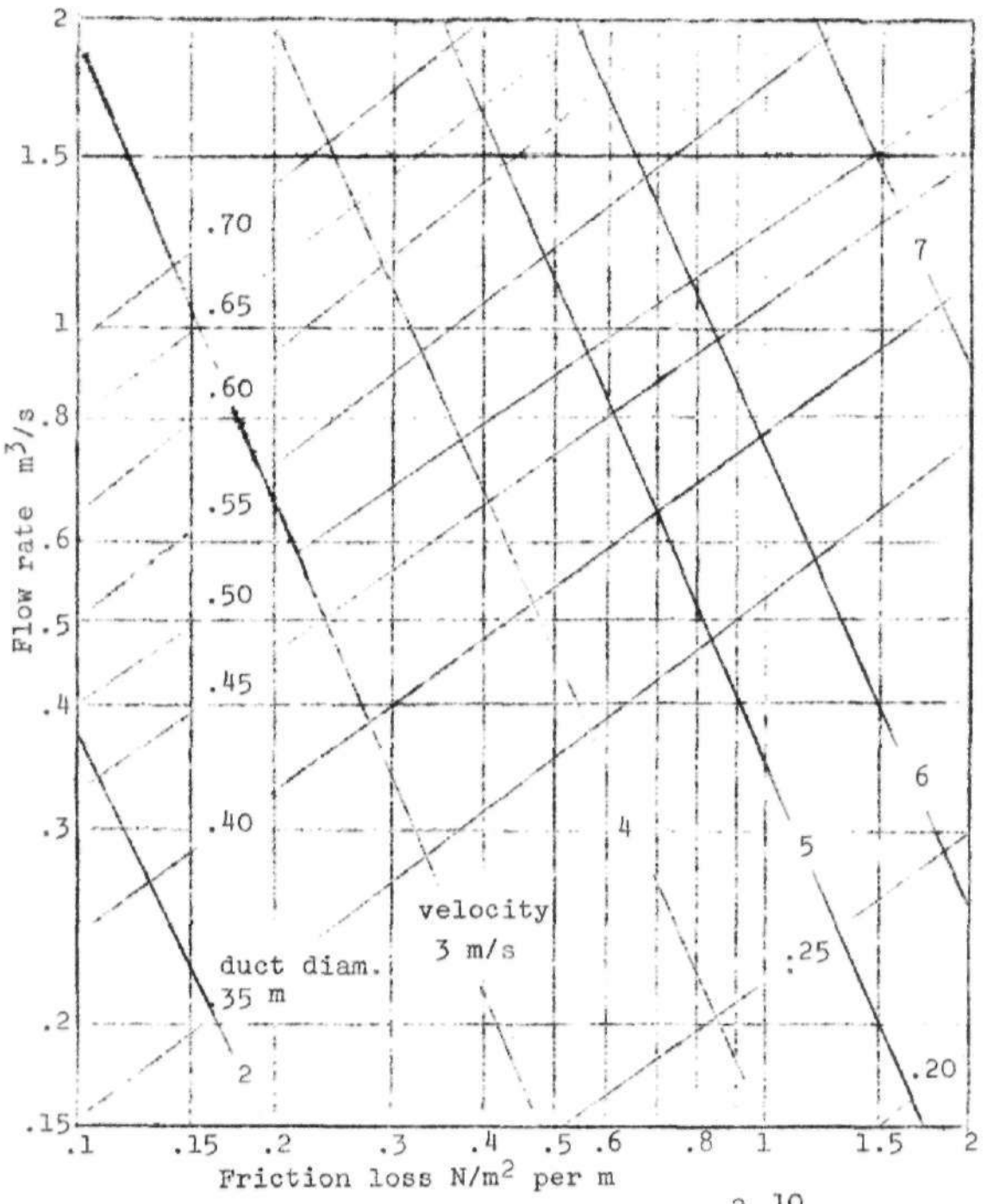


Fig. 9.1 Friction loss in duct (N/m^2) [10]

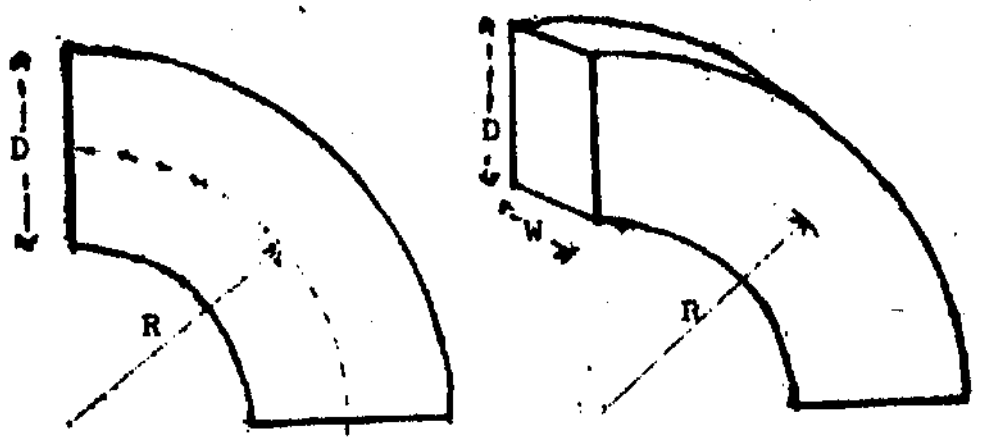
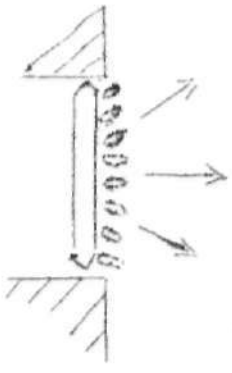
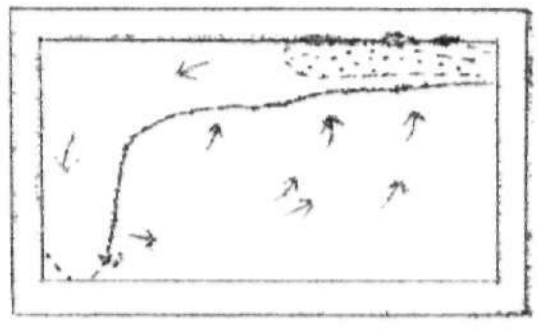


Fig. 9.2 Round and rectangular elbow





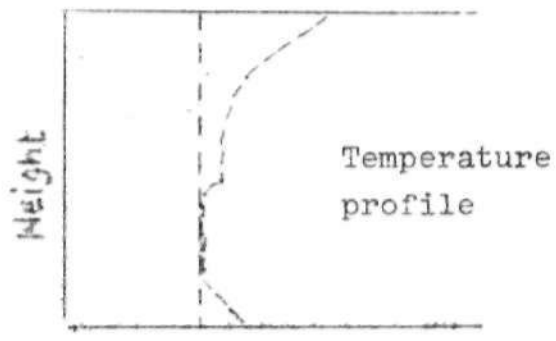


Fig. 9.3 Single side wall outlet

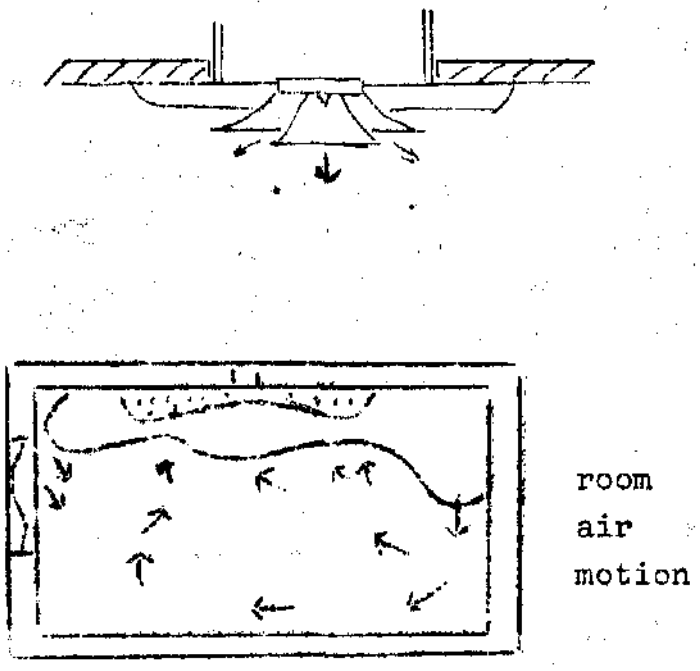
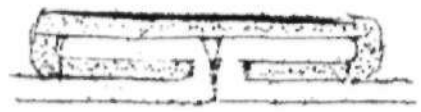
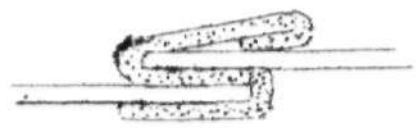


Fig. 9.4 Single ceiling diffuser



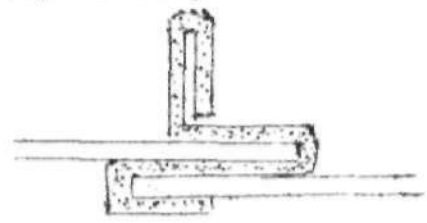
a) drive slip



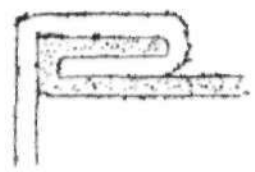
b) S-slip



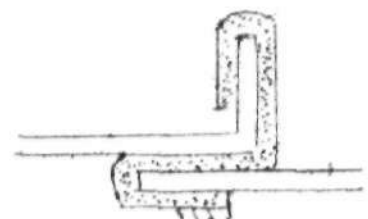
c) inside groove seam



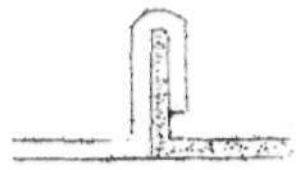
d) bar slip



e) sliding seam



f) pocket joint section at clip punch



g) standing seam



h) Pittsburgh seam

Fig. 9.5 Methods of duct construction

CHAPTER 10

LAYOUT AND SIZING OF DUCTS

10.1 Supply duct layout

A variety of physical, aesthetic, and performance factors affect the choice of outlet locations. The air-distribution relations concerning throw, drop and room circulation are primary considerations. Local loads due to concentration of people, equipment heat, outside wall and glass surfaces, and open doors may modify the choice of location.

The convective heat loss from ceiling light results in stratified warm air at the ceiling. With excessive light loads, a gravity or positive exhaust at ceiling level may be needed. Construction of the building will necessarily influence the layout of the air distribution system. However desirable it may be to locate an outlet in a given position, the physical limitations of walls, beams, columns and decorative treatment may prevent the use of such location. In addition, the feasibility of running the duct to the desired location should be considered.

The accepted procedure for locating outlets is as follows:

a) Ceiling diffusers

Where the duct is concealed above the ceiling, some type of ceiling diffuser should be used. Where the duct is under the ceiling, exposed or furred, either side wall or

ceiling outlets can be used. Down-blow outlets, where used, should be selected to complete the mixing of the air stream before it enters the occupied zone. High ceilings, as in banks or theatres, may permit the use of small jets directed downward at low or medium pressures.

b) Wall grilles

High location is preferable where a ceiling is free from obstructions. Grilles should not be too near the ceiling (preferably at least 300 mm down); otherwise dirt streaking may result. Where beams are encountered, a lower position is recommended in order to obtain a horizontal or slightly upward blow. A low location for wall grilles is not considered satisfactory unless the air is directed upward at a steep angle in such a manner that it does not strike the occupants or furniture directly.

Side wall grilles are selected to supply conditioned air to the rooms. The number of outlets required in a given space will depend upon the spread of an outlet. Air will not be thrown at as great an angle as is indicated by the vane setting since there is an angle loss of 10° to 20°, depending on the amount of turn given to the air and on the vane ratio.

A straight vane setting produces an included angle in both planes averaging 14° to 24°; when the end vanes are at 45° angle, and all other vanes set at intermediate angles to give a fanning effect, will produce an air stream with a horizontal included angle of approximately 60°;

under this condition the throw is about halved. Fig. 10.1 shows the air supply duct layout for the two floors.

10.2 Supply air quantities

The supply air quantities required for each room is shown in Table 7.4. This is divided by the number of grilles in each room to obtain flow rate as shown in Fig. 10.2.

10.2.1. Correction of air quantities

Heat is absorbed outside the conditioned space, resulting in a waste of refrigeration because of the cooling of the unconditioned space. This increases the compressor and dehumidifier load as well as the air quantities, and must be accounted for on the cooling load estimate. The air quantity must be increased at each grille to compensate for the lower temperature head from the grille to theroom due to temperature rise of the air in the duct.

Where the supply duct runs through unconditioned space, the air quantity should be increased to offset loss or gain.This adjustment is obtained as a correction factor (CF) to the air quantity equal to [5]

$$CF = \frac{1}{1 - (\Delta t_{rise} / \Delta t_{design})} \qquad (10\text{-}1)$$

where Δt_{rise} is the temperature rise in the duct

Δt_{design} is the temperature difference between room andsupply air.

Δt_{rise} is estimated from Table 10.1, which shows temperature rise in degrees Celcius per 10m length of duct for $10^{o}C$ differential between inside and outside of the duct. The values are based on velocities ranging from 3 to 6 m/s, duct with aespect ratio 2. The overall transmission factor is taken at 5 m/s air velocity as 1.25 for uninsulated duct and 0.25 for insulated ducts.

The correction factors (CF) are calculated as shown in Table 10.2.

10.2.2. Redistribution of air quantities

When the duct runs through conditioned space, it tends to cool the room this does not require more air supplied but does require a redistribution tothe various grilles. Such redistribution is necessary where the duct is long and where a multiplicity of rooms are served from the same run of ductwork.

The redistribution of air quantities is obtained by subtracting from the grille the flow rate equivalent of cooling produced by the duct[5]

$$Q' = \frac{Q \times \text{Temperature rise in duct}}{\text{Corrected } \Delta t_{design}} \qquad (10\text{-}2)$$

Fig.10.3 shows the temperature distribution in the ducts, needed in the calculation of temperature rise in the duct in the conditioned space.

The redistributedgrille air quantities are as shown in Table 10.3.

10.3 Sizing of supply air ducts

A low velocity duct system is used for the supply air duct, and the sstatic regain method employed to size the ducts.

The general procedure for use of the static regain method is to first select a velocity for the duct attached to the fan or supply plenum. With the capacity, this establishes the size of this main duct. The run of duct that appears to have the largest flow resistance is then designed first, in this case section 1. A velocity is assumed for the next section in the run and the static pressure regain is compared to the lost pressure for that section.

Assume a rectangular duct of dimension 600mm X 300mm.

D_1 = 0.404 m

A_1 = 0.128 m^2

V_1 = Q_1/A_1 = 1.0254/0.128 m/s

= 8.00 m/s

h_{v_1} = $(V_1/1.29)^2$ N/m^2

= 38.45 N/m^2

The loss coefficient for a formed entrance is 0.03

$$h_{c1} = C\, h_{v_1} + h'_o\, L \quad N/m^2 \qquad (10\text{-}3)$$

where h'_o is loss due to friction per m of duct

$$h'_{o_1} = 0.03 \times 38.45 + 1.60 \times 10.46 \ N/m^2$$
$$= 20.57 \ N/m^2$$

The total pressure required at the plenum to meet the needs of section 1 is then given by

$$h_1 = h_{o_1} + h_{v_1} \qquad (10\text{-}4)$$
$$= 20.57 + 38.45 \ N/m^2$$
$$= 59.02 \ N/m^2$$

If the static regain method could be followed exactly, this would be the system total pressure, exclusive of the terminal box requirement, because velocity pressure is to be converted to static pressure to offset the lost pressure.

To size section 2, it is required that the increase in static pressure (decrease in velocity pressure) in the transition must equal the lost pressure in the remaining part of section 2.

This is equivalent to

$$h_1 - h_2 = (h_{v_2} - h_{v_1}) + h_t + h_{o_2}$$
$$= 0 \qquad (10\text{-}5)$$

or may be generalised as

$$h_u - h_d = (h_{v_d} - h_{v_u}) + \sum (h_o) = 0 \qquad (10\text{-}6)$$

where subscripts u and d refer to upstream and downstream respectively.

Assume a diameter of 0.37 m for section 2

D_2 = 0.370 m

V_2 = 0.703/0.107 m/s
= 6.54 m/s

h_{v_2} = 25.70 N/m^2

h'_o = 1.5 N/m^2 per m C_t = 0.1

h_t = 0.1 X 25.70 N/m^2
= 2.57 N/m^2

h_{o_2} = 1.5 X 6.64 N/m^2
= 9.96 N/m^2

$h_1 - h_2$ = (25.70 - 38.45) + 2.57 + 9.96 N/m^2
= - 0.22 N/m^2
$\neq$ 0

This indicates that a larger increase in static pressure than required was achieved in the transition. Hence, assume a smaller diameter until

$h_1 - h_2$ 0.10 N/m^2

A diameter of D_2= 0.369 m gives a value of .04 N/m^2 for $(h_1 - h_2)$, hence the diameter of 0.369 m is chosen for section 2, this and subsequent calculations are shown in a tabular form in Table 10.4.

The supply grilles are selected to give the designed flow rate and throw, and to have approximately the same static pressure drop (30 N/m^2). The selections are made from performance data of supply grille manufacturers. The following grille sizes, shown in" Table 10.5, are selected from data by Environmental Elements Corporation, Dallas, Texas.

10.4 Return air duct system

The return air intake generally has very little effect on the room air motion. However the location may have considerable effect on the performance of the cooling equipment. Since it is desirable to returnthe warmest air to the cooling coil, the retun air intakes should be located in a stagnant region.

Selection and location ofreturn grilles is based on the following:

a) Velocity

Air approaches the returngrille from all directions, and the velocity falls off veryrapidly as the distance from the grille increases. The general drift of the return air towards the grille should not be higher than 0.1 m/s. If a single return grill satisfies this criterion there is no advantage in having multiple grilles.

b) Allowable pressure drop

Return air grilles are quite varied in design. The construction of the grille has very little to do with the overall performance of the system except to introduce some pressure loss and noise if not properly sized. The appearance of a return grille is important, and the louvre design is usually based on this. The return air grilles are also selected from performance data of Environmental Elements Corporation.

c) Noise

This problem is similar to that of the supply outlets, except that return grille areas are larger and usually nearer ear level. Grille velocities should not exceed 75 per cent of those for supply outlets.

The return air duct system, as shown in Fig. 10.4, is sized by the velocity reduction method.

The return air amounts to 2.09 m^3/s and thus the remaining 0.29 m^3/s (10 per cent of supply air) is made up from outside air through a grille of area .150 m^2 (460 x 360 mm) and having a pressure drop of 17 N/m^2.

Hence the total pressure requirement, excluding that of the air handling unit, will be approximately

return air system	17 N/m^2
supply duct system	50 N/m^2
and supply grilles	30 N/m^2
	97 N/m^2

This would require a class I fan in the air handler.

Table 10.1 Temperature rise Δt_{rise} [5]

Flow rate Q m^3/s	Δt_{rise} ($^oC/m^oC$) X 100 Uninsulated	25mm Cork insulation	7mm
0.0944	2.42	0.88	1.54
0.2832	1.65	0.46	1.19
0.4719	1.36	0.34	1.02
0.6607	1.18	0.28	0.90
0.9438	1.00	0.23	0.77
1.4158	0.83	0.19	0.64
1.8877	0.71	0.17	0.54

Table 10.2 Correction factors (CF)

Q m^3/s	Δt_{rise} $^oC/m^oC$ x100	L m	Δt_{design} ^oC	Δt_{rise} ^oC	Total Δt_{rise} ^oC	CF
1.0049	0.21	4.80	24.50	0.25	0.25	
0.6888	0.26	1.00	24.25	0.06	0.31	
0.6888	0.92	3.05	20.19	0.57	0.88	1.06
0.5709	0.95	6.05	19.62	1.13	2.01	1.16
0.4576	1.06	6.05	18.49	1.18	3.19	1.28
0.3443	1.13	6.05	17.31	1.18	4.37	1.43
0.2867	1.19	4.23	16.13	0.81	5.18	1.56
0.1912	1.41	3.13	15.32	0.68	5.86	1.68
0.0956	1.54	3.13	14.64	0.71	6.57	1.83
0.3161	0.44	3.50	24.25	0.37	0.62	
0.3161	1.15	6.00	23.88	1.65	2.27	1.19
0.1581	1.44	6.10	22.23	1.95	4.22	1.41
1.2877	0.21	2.00	24.50	0.10	0.10	
0.9447	0.23	1.00	24.40	0.06	0.16	
0.9447	0.77	3.80	20.34	0.60	0.76	1.06
0.7557	0.92	7.60	19.74	1.38	2.14	1.17
0.5667	0.95	7.60	18.36	1.33	3.47	1.31
0.3778	1.10	7.60	17.03	1.42	4.89	1.51
0.1889	1.42	7.60	15.61	1.68	6.57	1.83
0.3430	1.48	3.50	24.40	2.36	2.46	
0.3430	1.13	3.05	18.14	0.63	3.09	1.27
0.1715	1.37	6.10	17.51	1.46	4.55	1.46

Table 10.3 Redistributed grille air quantities

Grille air quantity m^3/s	Duct CF	Corr.gr. m^3/s	Δt_{design} °C	Δt_{rise} °C	Q' m^3/s	Redistr. Q m^3/s
	0.6888		11.62	0.57	0.0337	
0.1179	1.06	0.1249				0.0605
	0.5709		10.49	1.13	0.0615	
0.1133	1.16	0.1314				0.0716
	0.4576		9.31	1.18	0.0580	
0.1133	1.28	0.1450				0.0911
	0.3443		8.13	1.18	0.0499	
0.0576	1.43	0.0824				0.0468
	0.2867		7.32	0.81	0.0317	
0.0955	1.56	0.1489				0.1183
	0.1912		6.64	0.68	0.0195	
0.0956	1.68	0.1606				0.1451
	0.0956		5.93	0.71	0.0114	
0.0956	1.83	0.1749				0.1692
	0.3161		10.23	1.65	0.0509	
0.0749	1.19	0.0891				0.0382
0.0831	1.19	0.0989				0.0803
	0.1581		8.28	1.95	0.0372	
0.0832	1.41	0.1173				0.0987
0.0749	1.41	0.1056				0.1056

Table 10.3 continued

	0.9447		11.74	0.60	0.0482
0.1890	1.06	0.2003			0.1018
	0.7557		10.36	1.38	0.1006
0.1890	1.17	0.2211			0.1291
	0.5667		9.03	1.33	0.0834
0.L889	1.31	0.2475			0.1706
	0.3778		7.61	1.42	0.0704
0.1889	1.51	0.2852			0.2233
	0.1889		5.93	1.68	0.0535
0.1889	1.83	0.3456			0.3189
	0.3430		9.41	0.63	0.0229
0.1715	1.27	0.2178			0.1792
	0.1715		7.95	1.46	0.0314
0.1715	1.46	0.2504			0.2347

Note: In cases where the duct between two outlets lies in two rooms, the air quantity subtracted from the grille is taken in proportion to the length of the duct.

Table 10.4 Duct sizing by static regain method

	Dimensions mm x mm	D m	Q m^3/s	V m/s	h_{v_2} N/m^2	h'_{o_2} $N/m^2/m$	L m	$\sum h_o$ N/m^2	$h_u - h_d$ N/m^2
1	460 x 300	.404	1.0254	8.00	38.45	1.60	10.46	20.57	
2	380 x 300	.369	.7026	6.57	25.94	1.50	6.63	12.54	.03
3	420 x 300	.383	.6421	5.57	18.67	.90	6.05	7.31	.04
4	440 x 300	.392	.5705	4.73	13.43	.65	6.05	5.28	.04
5	440 x 300	.392	.4794	3.97	9.48	.50	6.05	3.97	.03
6	460 x 300	.407	.4326	3.33	6.64	.32	7.08	2.93	.09
7	380 x 300	.366	.3143	2.99	5.36	.22	3.13	1.22	-.06
8	230 x 300	.286	.1692	2.63	4.17	.25	3.13	1.20	.01
9	210 x 300	.273	.3228	5.51	18.27	1.20	15.23	20.11	-.07
10	170 x 300	.243	.2043	4.41	11.66	.90	6.10	6.66	.04
1'	370 x 500	.470	1.3576	7.83	36.80	1.50	2.00	6.68	
2'	310 x 500	.426	.9437	6.62	26.34	1.00	7.78	10.42	-.04
3'	330 x 500	.440	.8419	5.54	13.42	.80	7.60	7.92	.00
4'	330 x 500	.442	.7128	4.65	12.97	.55	7.60	5.48	.02
5'	310 x 500	.425	.5422	3.82	8.78	.43	7.60	4.15	-.05
6'	230 x 500	.363	.3189	3.08	5.71	.33	7.60	3.08	.01
7'	150 x 500	.287	.4139	6.40	24.60	1.50	6.55	12.28	.08
8'	110 x 500	.242	.2347	5.10	15.65	1.20	6.10	8.88	-.07

Table 10.5 Supply grille sizes

Grille flowrate m^3/s	Grille dimensions mm
0.0605	180 x 150
0.0716	180 x 150
0.0911	180 x 150
0.0468	150 x 150
0.1183	255 x 150
0.1451	360 x 150
0.1692	360 x 150
0.0382	180 x 150
0.0803	180 x 150
0.0987	305 x 150
0.1056	305 x 150
0.1018	610 x 205
0.1291	610 x 205
0.1706	610 x 205
0.2233	610 x 205
0.3189	760 x 205
0.1792	305 x 205
0.2347	305 x 205

Table 10.6 Recommended return air grille velocities[10]

Grille location	velocity m/s (gross area)
Above occupied zone	4.0 +
Within occupied zone	
not near seats	3.0 - 4.0
near seats	2.0 - 3.0
Door or wall louvres	1.0 - 1.5
Undercutting of doors through undercut area	1.0 - 1.5

Table 10.7 Sizing of return air duct system

	Duct or Grille diam. m	dimen.mm	NC	flowrate m^3/s	V m/s	L m	h'_o	h_o N/m^2	Σh_o
1	.798	800 x 660		1.75	3.5	19.39	.15	2.91	
2	.602	800 x 360		0.91	3.2	14.02	.20	2.81	
3	.363	300 x 375		0.31	3.0	4.50	.30	1.35	
4	.271	300 x 205		0.15	2.6	18.69	.32	5.98	
IV		610 x 255	-	0.15				2.74	15.79
5	.597	500 x 600		0.84	3.0	12.65	.18	2.28	
II		660 x 405	21	0.44				10.15	15.34
6	.443	500 x 330		0.40	2.6	12.65	.16	2.02	
III		660 x 405	18	0.40				8.46	15.67
VI		760 x 255	19	0.60				11.11	16.83
7	.265	300 x 200		0.16	2.9	4.80	.45	2.16	
V		915 x 150	11	0.16				5.64	14.87
8	.352	560 x 220		0.34	3.0	29.58	.30	8.87	
I		610 x 205	16	0.34				7.87	16.74

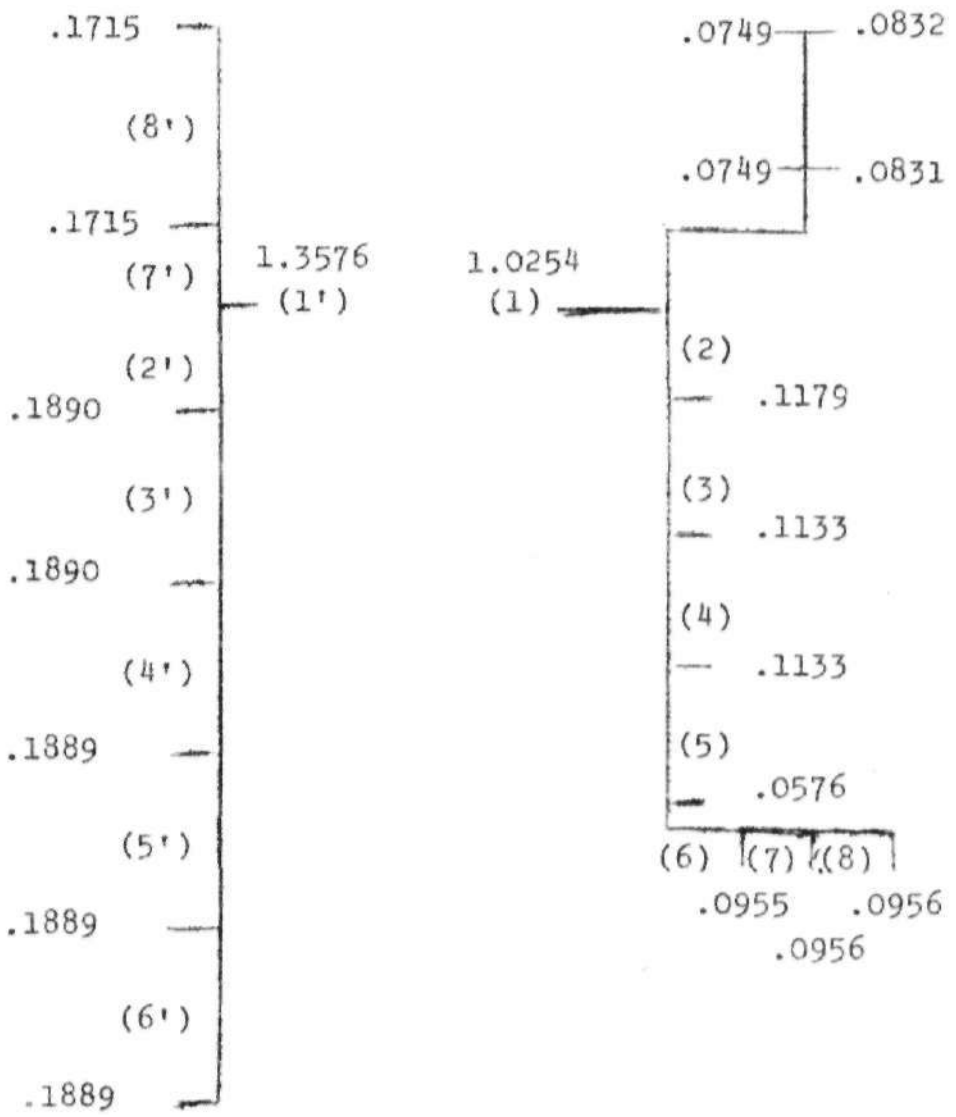


Fig. 10.2 Grille air quantities (m^3/s)

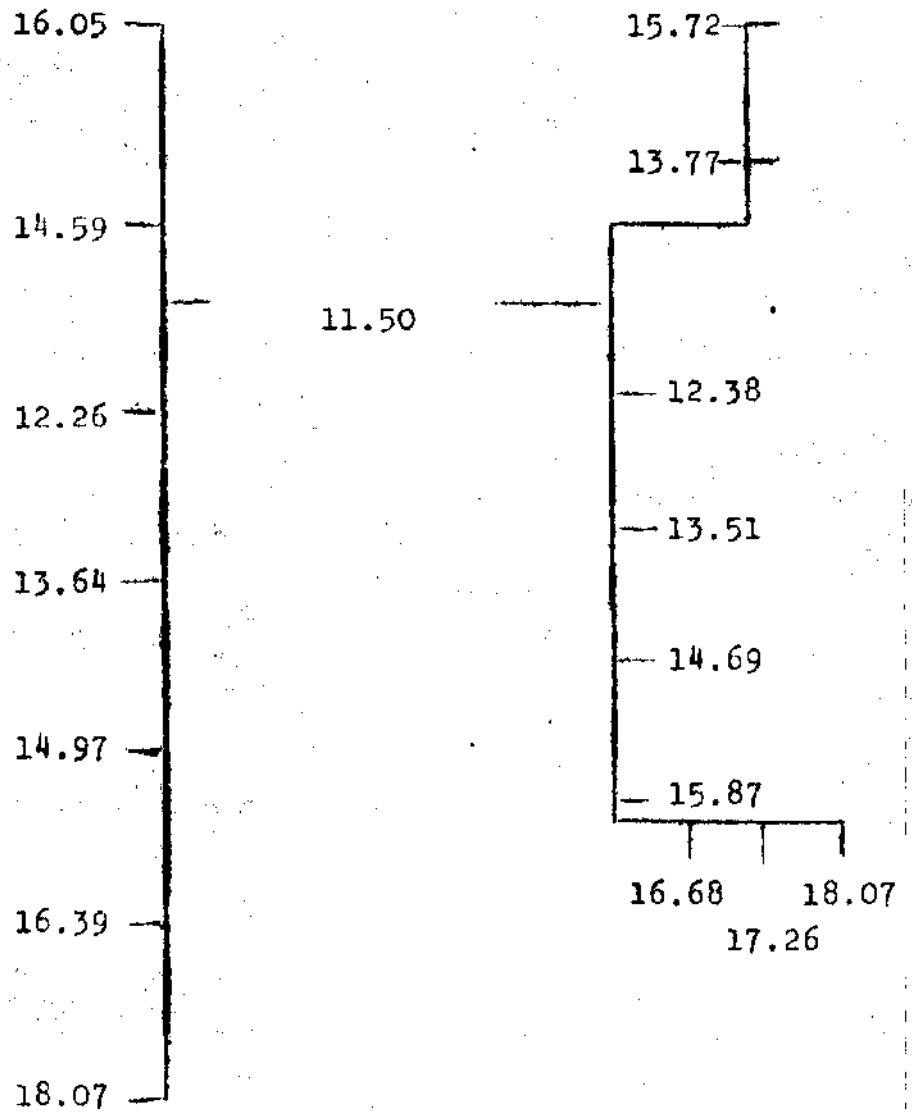


Fig. 10.3 Temperature distribution in the duct

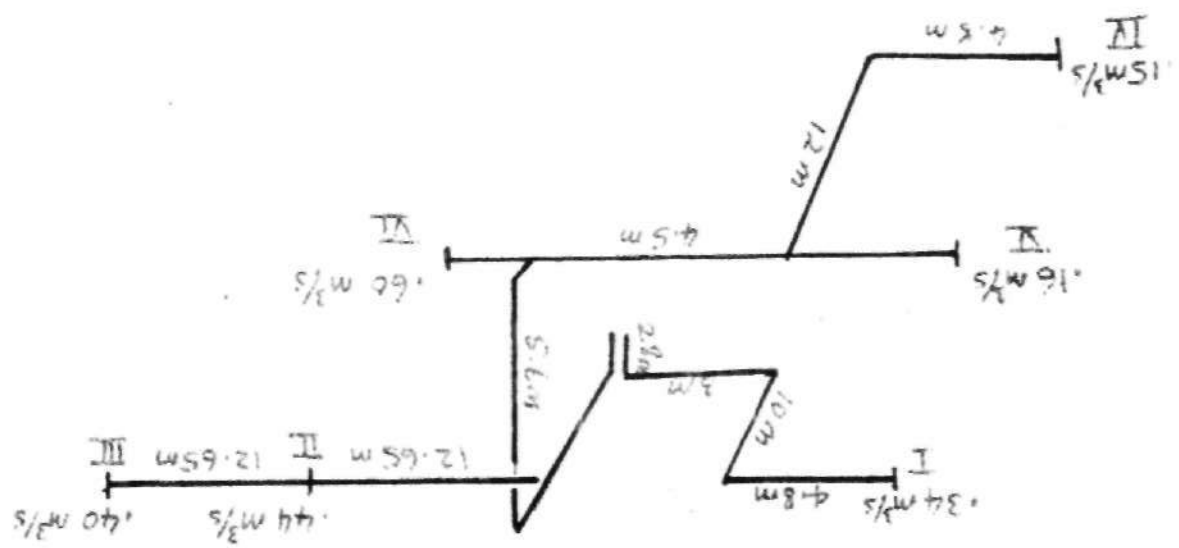


Fig. 10.4 Return air system

CONCLUSION

The second and the third floors of the Mechanical Engineering Department building may be air conditioned using either a number of small self-contained room units or a central all air single zone system. Using an air-cooled condensing unit (Westinghouse ST 020) and an air handling unit (Westinghouse AB 080) a saving of ₦550 per annum can be realized, in addition to having better air motion within the rooms, as compared to a number of room units that will be required to condition the building.

The beamed structure of the ceiling makes the use of suspended ceiling unsuitable, because it would reduce the ceiling height considerably. Hence exposed ductwork with side wall outlets are used to supply air to the conditioned space. Therefore rectangular ducts are designed keeping the vertical dimensions constant, 300 mm for the second floor and 500 mm for the third.

The supply duct is designed as a low velocity system with total pressure of about 200 N/m^2 and the ducts may be field fabricated.

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